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BOILERS
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BY

HUBERT E. COLLINS

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BOILERS

INTRODUCTION

This volume endeavors to furnish the reader with much new and valuable material on an old subject, together with much standard information which every engineer likes to have at his hand. A glance at the chapter headings will show the scope of the book. It will be seen that the subject is pretty fully covered from the working conditions inside of a boiler to simple talks on the various phases of boiler practice. It also covers the design of boiler furnaces for wood burning, and much other useful material.

One very important feature is the portion on the safety valve based on Mr. Fred R. Low's supplement to Power on that subject. The author is indebted to Mr. Low for permission to incorporate this material in the book, and to various other contributors, whose articles have been used as a whole or in part in the work.

Hubert E. Collins.

New York, *November*, 1908.

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I

WATCHING A BOILER AT WORK[1]

If we take a test-tube filled with water nearly to the top and hold it over a Bunsen flame, the water boils violently and overflows the tube. This violent over-boiling is due to the conflicting action of the ascending and descending currents of steam and water in the tube. On the other hand, if we take a tube shaped like a U, the arms of which are connected together at the top, fill it with water and place one leg of the U in the flame, a direct circulation soon commences. The water passes along in one direction and the steam is liberated at the surface. In this case there is very little violent ebullition, because there are no counter currents and the steam is discharged quietly over a liberal surface.

Desiring to ascertain just how nearly a boiler could be designed to work upon the U-tube principle of circulation, after several trials the model boiler shown in Fig. 1 was produced.

This model was built entirely of brass. It contained three drums four inches in diameter and 120 brass tubes one-quarter of an inch in diameter. The tubes were connected into headers and into the circum-

[1] Contributed to Power by C. Hill Smith.

ferences of the drums. The heads of all three drums were made of plate glass, for observation of the interior of the boiler when making steam. It will be noted that the design of the boiler closely resembles

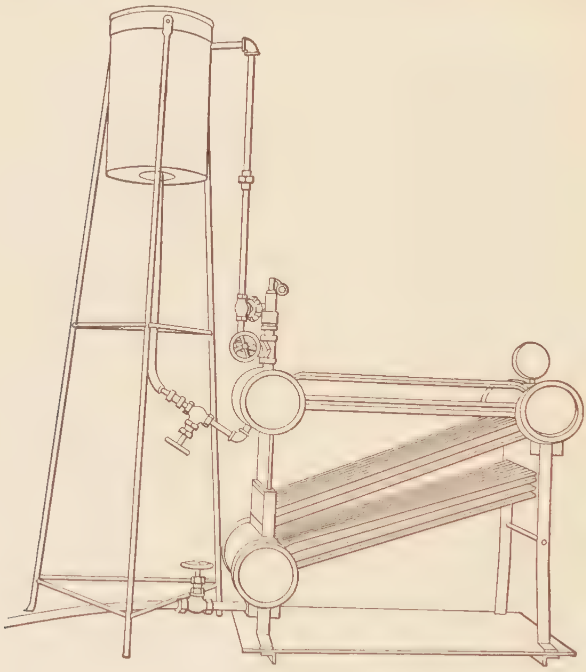

FIG. 1.

the U shape, only that one leg is considerably longer than the other, and there are two legs on the side of the U where the heat is applied.

Each of the three drums serves a special function

which will be noted from the description of the experiments. The two legs, instead of being connected together at the top, as was the case in the U-tube, are connected by two separate passages, one for the water to pass through and the other for the steam.

In preparing for the tests the boiler was mounted on a stand, so that the tubes inclined from the horizontal 20 degrees, and the whole was enclosed on all sides by brass plates. Alcohol lamps were placed inside the casing at a point to correspond with the regular location of grates, or at about one-fourth of the distance between the front headers and the rear drum. The steam outlet was located on the rear drum, as was the safety valve, for experimental reasons, although in actual practice the safety valve would be located on the front drum. The feed-pipe was introduced in the rear drum, while the blow-off entered the lowest point of the lower drum, which we will call the mud-drum. The boiler was attached to an open condenser.

The boiler being ready for test, it was filled with cold water until the upper drums were filled to one-half their volume. Candles were placed behind one head of each of the three drums for the purpose of lighting the inside. The alcohol lamps were then lighted and the boiler interior was ready to observe through the glass heads of the drums.

The first action noted was in the front drum, which served as a discharge chamber for all the steaming tubes. The tubes of the lower bank discharged into it through headers, while those of the upper bank dis-

charged into it independently. Many faint, oily-white streamers were seen to rise from the nipples connecting the headers to the front drum, passing upward to the surface of the drum. They resembled little streamers of white smoke. On reaching the surface of the water they passed into the horizontal circulating tubes which connected the two upper drums. These little streamers were heated water, which, being lighter than the water in the drum, rose to the surface. This same action soon appeared from the ends of the upper bank of tubes, the little streamers rising in a similar manner and passing into the horizontal tubes.

By observing the rear drum, the little streamers could be seen coming into this drum from the front drum. Here they turned downward into the vertical tubes which connected the rear drum to the rear headers and the mud-drum. No action could be noted in the mud-drum, which fact seemed to indicate that these currents of water passed into the upper tubes and thence into the front drum again, as the action from these tubes appeared very much more decided than the action from the nipples, notwithstanding the fact that the lower tubes were nearer to the flame than the upper ones. This was undoubtedly due to the fact that the heated currents of water remained as near the surface as possible, while the colder water passed to the bottom of the boiler, having greater specific gravity.

Particles of sediment could be seen coming down the vertical circulating tubes into the mud-drum, evidently precipitated from the water that was being

heated. This sediment passed to the bottom of the drum, where it remained. A very gradual action was now noted in the mud-drum in the nature of similar currents of water coming down the vertical tubes. These currents acted strangely on entering the drum; they spread out on coming in contact with the colder and denser water lying at the bottom. By placing the finger on the upper portion of the glass head and then on the lower, quite a difference in temperature was noted. Little streamers of heated water soon commenced to pass into the lower bank of streaming tubes which were connected into this drum. They passed across the drum with a sort of shivering motion. A new and very interesting phenomenon was now noticed. Occasionally there would appear from the ends of the steaming tubes little rings of heated water, which shot across the drum with considerable velocity.

The action in the front drum became very much more pronounced and air bubbles appeared from the nipples and tubes. The boiler was circulating water with great rapidity in the same direction and it was noticed, by placing the hand on different parts of the boiler, that all parts were of the same temperature. The air bubbles now discharged in great quantities from the tubes and nipples and rising to the surface disturbed the water level considerably. It was noted that they floated along under the surface of the water before they broke.

Gradually these air bubbles ceased to appear and a new kind took their places. The latter were steam bubbles and they discharged into the drum with greater

velocity than the former. On reaching the surface of the water they broke immediately, but they agitated the water level to a much greater extent. Fountains of water would shoot up into the drum for quite a distance and showed very vividly the conditions present in the shell type of boiler, where there are no defined paths for the water and steam to travel and nothing to prevent their conflict with each other. This also shows the cause for wet steam, and the great danger of entraining water with steam, as is the case where the steam is removed from the same place where violent ebullition is present.

While the water level in the front drum was violently agitated, the water level in the rear drum remained perfectly calm. No steam was generated in this drum, as the horizontal tubes connecting it to the front drum only circulated water that had been freed of its steam.

As the steam gage soon registered a pressure of 9 pounds, the main stop-valve was opened to allow the steam to flow to the condenser. The abrupt release of pressure caused the water to expand suddenly and the water level rose about one-quarter of an inch. This was evidently due to the sudden generation of steam caused by the drop in the pressure. This increased ebullition caused a very violent action in the front drum and the circulation of water through the boiler increased greatly in velocity. The nipples and tubes in the front drum discharged great quantities of bubbles. The water level in the rear drum during this increase in ebullition showed but a few ripples,

which were evidently due to the vibration of the steam passing into the steam main, or the discharge into the water of the open condenser.

The sudden generation of steam caused by the opening of the steam valve and subsequent reduction in pressure, it is believed, explains how the partial rupture of the shell of a return-tubular boiler is advanced to a disastrous explosion by the unexpected increased generation of steam due to the lowered pressure.

The steam main was now closed sufficiently to allow the boiler to operate on a constant pressure of about 6 pounds. It operated very smoothly under these conditions and made a very interesting sight with the steam generating in the front drum, where the nipples and tubes discharged great quantities of bubbles.

The action in the mud-drum had in the meantime become well worth watching. In the other two drums the water showed clear in the candle light, but the color of the water in the mud-drum was very murky. Particles of sediment were noted settling to the bottom. The withdrawal of water from this drum by the steaming tubes did not appear to draw this sediment into the tubes, as the drum was of ample size so that the suction was not felt at the bottom where the sediment deposited. This emphasized clearly the advantage of a very large mud-drum to allow of the thorough settling of the sediment.

The condition of the front drum was thought to be too violent for good practice, because the ebullition indicated restriction of circulation. The boiler was

put out of operation for the purpose of making changes in this drum to prevent extreme ebullition. The glass heads were removed and other nipples inserted over the nipples that connected the headers into the drum, it being here that the most violent discharge of steam was discernible. These new nipples were cut long enough to reach to the water level, or just a trifle below it.

The glass heads were replaced and the boiler put in operation again. The circulation was similar to that in the first test, and no real difference was noted until the boiler commenced to make steam. Then it was seen that the ebullition in this drum was considerably reduced, the agitation that remained being caused by the discharge from the tubes of the upper bank. This reduction was evidently due to having provided a channel through which the water and steam from the nipples might flow to the surface of the water and so prevent contact with the water in the drum. As the steam and water no longer had to force their way to the surface, the disturbance of the water level was naturally reduced entirely in this direction. The water rose from the nipples in little fountains, the steam disengaging from it in the upper part of the drum.

The boiler was operated under very severe conditions to try the value of this addition of nipples. The main stop-valve was suddenly opened after a considerable steam pressure was obtained. It had very little effect on the water level in this drum, only causing the nipples to discharge fountains of water quite a distance into the drum. No water was thrown into

the superheating tubes, as the fountains of water discharged vertically and fell back immediately to the water level.

The value of this attachment being proved, the boiler was blown down, and after the water was all withdrawn from the boiler considerable sediment was found in the bottom of the lower or mud-drum. Nowhere else was sediment found, as the drums offered no opportunities for the sediment to settle, being pierced at their lowest points by tubes and nipples. The tubes were inclined 20 degrees, which insured thorough draining of the boiler.

From the foregoing experiments many points of great value for improvement in design of water-tube boilers can be derived. The violent ebullition in the front drum shows conclusively that steam should not be withdrawn from the boiler at a point where ebullition is present, on account of the danger of getting water entrained with the steam. It also shows that any sudden reduction of the pressure causes violent ebullition and priming. The front-drum conditions show that this is a good place to locate the safety valve, as the sudden opening of it would cause no liability of priming if the steam is not withdrawn from this drum.

The total lack of any ebullition in the rear drum shows that this is an ideal spot to remove the steam. It was noted that, owing to the large amount of separating surface provided, the opening of the steam valve caused no priming in this drum. Another feature to be noted is the value of a large mud-drum to

provide ample opportunity for the sediment to settle, and also to provide a large supply of water for the bottom tubes. It would be impossible to force the boiler hard enough to drain this drum of water, so the danger of burning out these tubes is eliminated.

The provision of the long nipples in the front drum proved the advantage of providing separate passages to allow the steam and water to reach the surface of the water, thus obviating the necessity of their forcing their way to the surface through the large body of water in this drum and so cause violent ebullition.

II

SIMPLE TALK ON EFFICIENCY OF RIVETED JOINTS

MATTER is conceived to be composed of myriads of tiny molecules separated from each other by distances which are very considerable as compared with their diameters, and held in fixed relation to each other in solid bodies, by such an attraction as holds the earth to the sun or the moon to the earth. When we tear a piece of boiler sheet apart it is the attraction of these molecules which we are overcoming, and if the metal is uniform the force required to separate it will depend upon the surface which we expose. It will take twice as much force to pull the larger of the two bars in Fig. 2 apart as it will the smaller, because there is twice as much surface exposed at B as at A, and the attraction of twice as many molecules to overcome.

The force tending to pull a body apart in this way is called a "tensile" force, and the resistance to the force necessary to pull a piece apart is called its "ultimate tensile strength." This is usually given in pounds per square inch, and is for boiler iron around 45,000 and for boiler steel around 60,000 pounds. It should be found stamped on the sheets of which boilers are made. Suppose we have a single riveted joint like Fig. 3. We

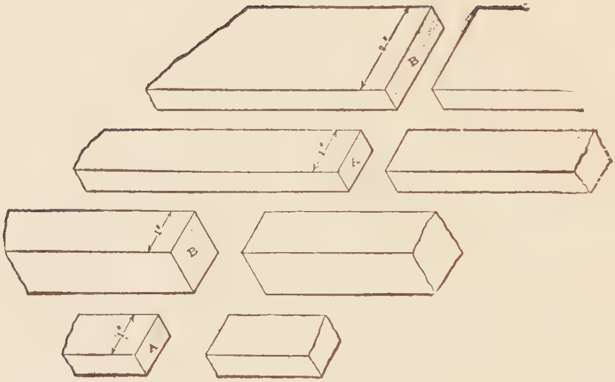

FIG. 2.

can divide it into strips as by the dotted lines half-way between the rivets, and consider one of these strips, for since they are all alike, what is true of one will be true of all. The width of each strip will be the same as

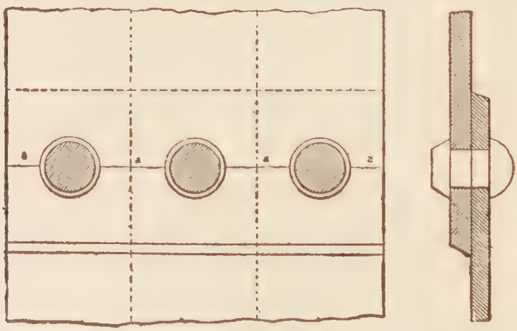

FIG. 3.

the distance from center to center of the rivets. This is called the "pitch." Let us suppose the pitch to be $2\frac{1}{4}$ inches, the diameter of the rivet 1 inch, the thickness of the plate $\frac{1}{2}$ inch, the tensile strength of the plate 60,000 and the shearing strength of the rivets 43,000 pounds.

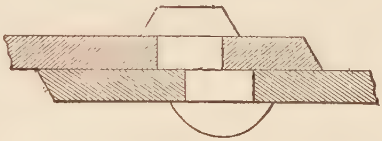

FIG. 4.

There are two ways in which this joint can fail: by tearing the sheet apart where there is the least of it to break, as at *a a a a*, Fig. 3, or by shearing the rivet as in Fig. 4.

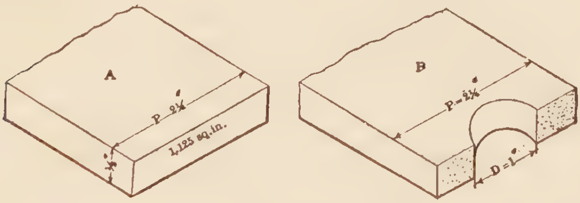

FIG. 5.

If the strip were whole as at *A* in Fig. 5, it would have

$$2\frac{1}{4} \times \frac{1}{2} = 1.125 \text{ square inches}$$

of section, and since it takes 60,000 pounds to pull one square inch apart it would take

$$1.125 \times 60{,}000 = 67{,}500 \text{ pounds}$$

to separate it.

But 1 inch of the sheet has been cut out for the rivet, so that there are left only

$$2\tfrac{1}{4} - 1 = 1\tfrac{1}{4} \text{ inches}$$

of width to be separated, and

$$1\tfrac{1}{4} \times \tfrac{1}{2} = 0.625 \text{ square inch}$$

of area. This would stand a pull of only

$$0.625 \times 60{,}000 = 37{,}500 \text{ pounds.}$$

Whether the joint will part by tearing the sheet or shearing the rivet depends, of course, on which is the stronger. The rivet has

$$1 \times 1 \times 0.7854 = 0.7854 \text{ square inch}$$

of area, and it takes 49,000 pounds to shear each square inch, so that it would take a pull of

$$0.7854 \times 43{,}000 = 33{,}772.2 \text{ pounds.}$$

to shear the rivet.

It is evident that the rivets would go, then, long before the plate, and that the strength of the joint would be

$$33{,}772.2 \div 67{,}500 = 0.50$$

or 50 per cent. of the strength of the full plate.

But we can add to the rivet strength without reducing the plate strength by putting in another row of rivets behind the first row. In Fig. 6 two rivets have to be sheared, doubling the rivet strength without reducing the plate strength, for the holes for these extra rivets do not reduce the plate section along any one line if there is space enough between the rows. In Fig. 7 the

sheet is no more apt to part along the line *a a a a* than it would be if the second row of rivets were not there, and no more likely to part on the line *b b b b* than on the other. Any strip of a width equal to the pitch will

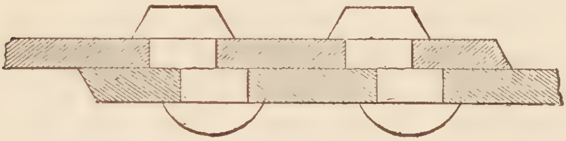

FIG. 6.

contain two rivets, whether we take it through the rivet centers, as at *A*, Fig. 7, or at equal distance to either side of one rivet in each row, as at *B* in the same figure. In the first case it includes one full rivet and two halves, and in the latter case two full rivets.

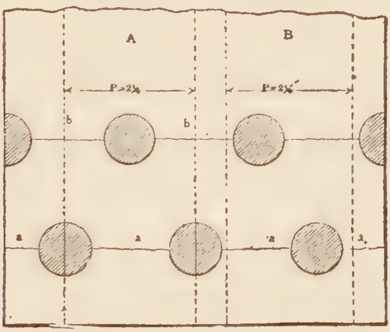

FIG. 7.

To find the efficiency of this joint, then, we calculate the efficiencies of the plate and use the lowest efficiency.

To calculate the plate efficiency, divide the difference

between the pitch and the diameter of the rivets by the pitch.

This is simpler than the operation which we went through above, which was

$$\frac{(\text{pitch-diam.}) \times \text{thickness} \times \text{tensile strength}}{\text{pitch} \times \text{thickness} \times \text{tensile strength}}$$

the numerator being the pull required to separate the sheet with the rivet holes cut out, and the denominator the pull required to separate the full sheet. As the thickness and tensile strength appear in both numerator and denominator, they cancel out.

To find the rivet efficiency, multiply the diameter of the rivet by itself, by 0.7854, *by the shearing strength per square inch and by the number of rows, and divide by the product of the pitch, thickness and tensile strength per square inch of section.*

These rules are applicable only to lap joints where the rivets are in single shear.

III

SIMPLE TALKS ON THE BURSTING STRENGTH OF BOILERS

There are two ways that a shell, such as is shown in the sketches herewith, might break under internal pressure. The sheets might tear lengthwise, letting the shell separate, as in Fig. 8, or they might tear across, letting it separate endwise, as in Fig. 9.

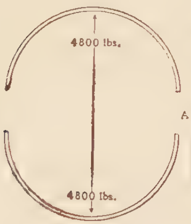

Fig. 8.

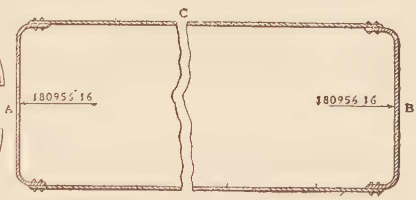

Fig. 9.

Which is it the more likely to do?

To push it apart endwise, as in Fig. 9, we have the force acting on the heads. This force is the pressure per square inch multiplied by the number of square inches in the head. The area of a circle is the diameter multiplied by itself and by 3.1416 and divided by 4; or since 3.1416 divided by 4 is .7854, the area is the square of the diameter multiplied by .7854.

Suppose the internal diameter of the shell to be 48 inches, and the pressure 100 pounds per square inch, the pressure on each head would be

$$48 \times 48 \times .7854 \times 100 = 180{,}956.16 \text{ pounds},$$

or over 90 tons. This pressure would act on each head, and the effect would be the same as though two weights of 180,956.16 pounds each were pulling against each other through the boiler, as in Fig. 10.

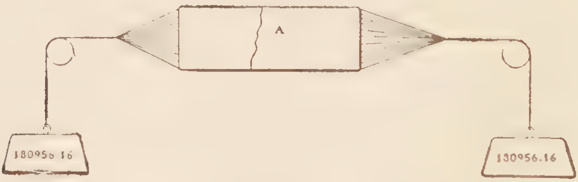

FIG. 10.

If the shell were not heavy enough to stand the strain, it would tear apart along the line where the metal happened to be the weakest, as at *A*. At first sight it looks as though the metal had to sustain both these forces or weights, and that the stress upon the shell would be twice 180,956.16 pounds; but a little consideration will show that this is not so. One simply furnishes the equal and opposite action with which every force must be resisted. A man pulling against a boy on a rope (Fig. 11) can pull no harder than the boy pulls against him. If he does he will pull the boy off his feet, and the strain on the rope will be only what one of them pulls, not the sum of both pulls. In order that the man may pull with a force of 50

pounds, the boy must hold against him with a force of 50 pounds. Both are pulling with a force of 50

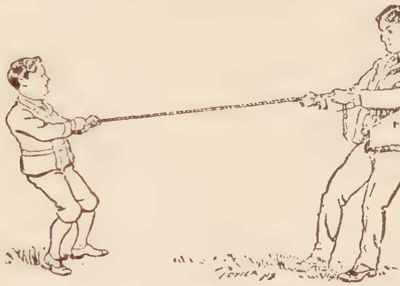

FIG. 11.

pounds, but the tension on the rope is 50 pounds, not 100. The boy might be replaced with a post (Fig. 12). Now, when the man pulls with a force of 50 pounds

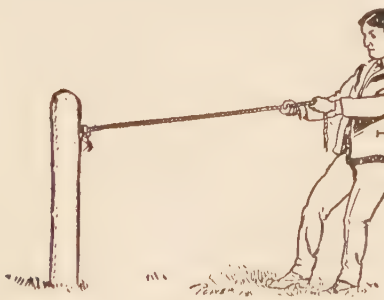

FIG. 12.

against the post, you would not say that there was 100 pounds tension on the rope; yet the post is pulling or holding against him with a force of 50 pounds

just as the boy did. In Fig. 13 it is easily seen that the tension on the cord is 50 pounds. You would not say that it was 100, if the pull of the weight were resisted by another weight of 50 pounds, as in Fig. 14, instead of by the floor.

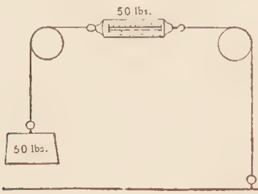

Fig. 13.

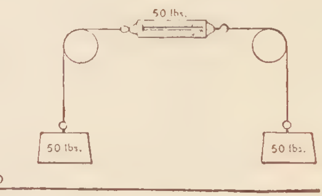

Fig. 14.

The shell is therefore in the case which we have imagined subjected to a force of 180,956.16 pounds, which tends to pull it apart endwise, as in Fig. 10.

To resist this there are as many running inches of shell as there are inches in the circumference.

The circumference is 3.1416 times the diameter, so that to pull the boiler in two

$$48 \times 3.1416 = 150.7968 \text{ inches}$$

of sheet would have to be pulled apart.

The force exerted upon each running inch of sheet would be the pressure acting endwise divided by the circumference, or

$$180{,}956.16 \div 150.7968 = 1{,}200 \text{ pounds.}$$

The area is

$$\frac{\text{diam.} \times \text{diam.} \times 3.1416}{4}$$

The circumference is

$$\text{Diam.} \times 3.1416.$$

Dividing the area by the circumference we have

$$\frac{\text{diam.} \times \text{diam.} \times 3.1416}{4 \times \text{diam.} \times 3.1416} = \frac{\text{diam.}}{4}$$

or the strain on each running inch of sheet per pound of pressure is one-fourth the diameter.

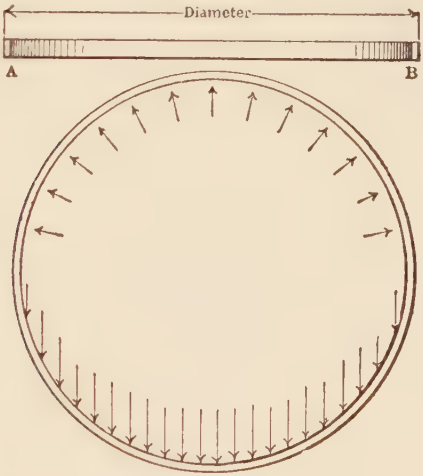

FIG. 15.

Now let us see what it would be in the other direction.

If we consider the pressure acting in all directions as in the upper half of Fig. 15, we should, to get the total pressure on the area, have to multiply the pres-

sure per square inch by the whole area, which would be the circumference for a strip 1 inch wide; but if we are considering the effect of pressure in one direction only, we must consider only the area in that direction. If we are studying the effect of the pressure in forcing the shell in the direction of the arrows in the lower half of Fig. 15, we must consider only the area which comes crosswise to that direction, the "projected area," as it is called; the area which the piece would present if we were to hold it up and look at it in the direction of the arrows or of the shadow which it would cast in rays of light running in the direction of the pressure. This, it will be easily recog-

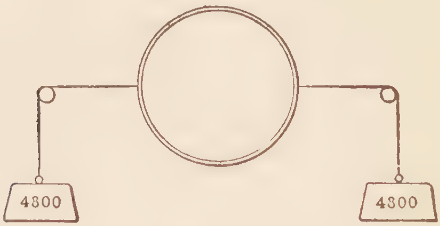

FIG. 16.

nized, is the diameter of the boiler wide and 1 inch high, as shown in Fig. 15, so that the number of square inches upon which the pressure is effective in one direction is equal to the diameter for a strip 1 inch wide. There is therefore a force tending to pull each 1-inch ring of the shell apart, as in Fig. 16, of $48 \times 100 = 4800$ pounds, and as this force is resisted by two running inches of metal, one at *A* and one at *B* (Fig. 15),

the stress per inch will be $4800 \div 2 = 2400$ pounds. This is just twice what we found it to be in the other direction; and it is plain that this should be so, for the stress per pound of pressure tending to burst the boiler, as in Fig. 8, is, as we have just seen,

$$\frac{\text{diam.}}{2}$$

which is just twice the $\frac{\text{diam.}}{4}$ which we found it to be in the other direction. It is for this reason that boilers are double riveted along the side or longitudinal seams, while single riveting is good enough for girth seams.

IV

SIMPLE TALKS ON THE BURSTING STRENGTH OF BOILERS

In the preceding chapter we found that a cylinder equally strong all over, will split lengthwise with one-half the pressure which would be needed to tear it apart endwise.

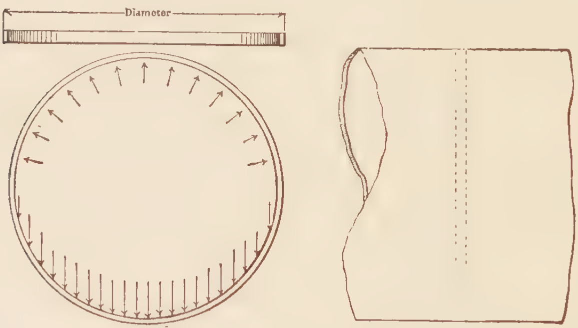

Fig. 17.

Let us see how much pressure it would take to burst a shell of this kind. We will consider a strip 1 inch in width, as in Fig. 17, for the action upon all the similar strips into which the boiler can be imagined to be spaced off will be the same. We see that the pressure

tending to pull the ring, 1 inch in width, apart is equal to the pressure per square inch multiplied by the diameter of the ring. The total pressure *in all directions*, acts on the circumference as shown by the radial arrows at the upper portion of the cut, but when we come to consider the force acting in *one* direction we must take the projected area in that direction; the area of the shadow, as explained before, cast by rays of light flowing in that direction, and that area would be that of the strip as we see it at the top of Fig. 17, 1 inch wide and the diameter of the boiler long.

It is sometimes hard for one to see why the diameter is used here, instead of the circumference, and a further illustration is here given.

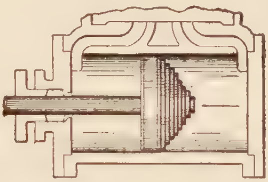

FIG. 18.

Suppose you had a piston in an engine cylinder made in steps like Fig. 18. This would have a good deal more surface to rust or to condense steam than would a flat piston, but it would have no more effective area for the production of power, would it? One hundred pounds behind it in the cylinder would push no harder on the crosshead with this than with a perfectly flat piston; because the sidewise pressure against

the steps is balanced by an equal pressure from the opposite side; only the pressure on the flat rings effective to move the piston forward, and the area of all these rings added together, is just the same as that of a flat surface of the same external diameter, as seen by the projection at the right.

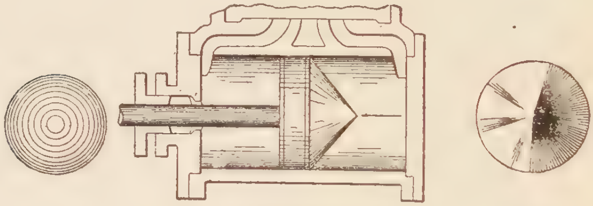

FIG. 19.

This would be just as true if the steps were a millionth or a hundred-millionth of an inch wide and high instead of an inch or more, so that it is just as true of a conical surface, like Fig. 19, as of Fig. 18, or of a concave

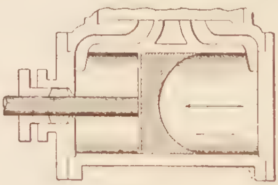

FIG. 20.

surface, like Fig. 20, as of either; and it is evident that it is the flattened-out area which one sees in looking at the object in the line of the force considered, the projected area in that direction as it is called, and not the real superficial area which is effective.

We have then a force equal to the pressure per square inch multiplied by the internal diameter of the shell tending to pull each inch in length of it apart, and we have *two* sections, *A* and *B*, Fig. 15, where the sheet must part.

The force tending to tear *each* of these is the pressure per square inch multiplied by the *radius*, or half the diameter of the shell. The resistance that the piece of shell will offer to being torn apart is the tensile strength per square inch multiplied by the number of square inches to be torn apart.

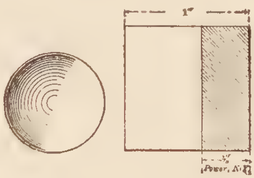

FIG. 21.

This area is one inch long and the thickness of the sheet in width. The area in square inches is therefore the same as the thickness in inches. If the plate were $\frac{3}{8}$ of an inch thick, for example, its section per inch of length would be $\frac{3}{8}$ of a square inch, as shown in Fig. 21.

The two opposing forces, which must be equal, not only at the point of fracture, but at all times, are:

Pressure per square inch $\times$ radius, and pull per square inch $\times$ thickness.

The pull on the sheet is called the tensile force. If we want to find the tensile force on the sheet for any pressure per square inch, we multiply that pressure by

the radius and divide by the thickness of the sheet in inches.

If we want to find the pressure per square inch necessary to get up a given tensile force per square inch, we multiply the given pull per square inch by the thickness of the plate and divide by the radius in inches.

We can find the pressure per square inch necessary to rupture the sheet by multiplying the ultimate tensile strength, that is, the tensile force required to pull a square inch of it apart by the thickness and dividing by the radius.

Example. — What pressure would be required to burst a tank 48 inches in diameter, made of steel $\frac{1}{4}$ of an inch in thickness, having a uniform tensile strength of 60,000 pounds per square inch?

$$\frac{\text{Tensile strength} \times \text{thickness}}{\text{radius}} = \text{pressure.}$$

$$\frac{60{,}000 \times .25}{24} = 625 \text{ lbs. per sq. in.}$$

But we cannot or do not in boiler practice get a shell of uniform strength. There have to be joints and these joints are not so strong as the plate itself. We will have a talk later about how to figure the strength of a riveted joint. Suppose the riveted joint was only 70 per cent. of the plate strength, then it would take only 70 per cent. of the force to pull it apart, and the result just found must be multiplied by .70 if the tank, instead of having a "uniform tensile strength of 60,000," has a sheet strength of 60,000 and a longitudinal seam of 70 per cent. efficiency.

The complete operation of finding the bursting strength of a boiler shell is

$$\frac{\text{Tensile strength} \times \text{thickness} \times \text{efficiency of joint}}{\text{radius}}.$$

RULE. — *Multiply the tensile strength of the weakest sheet in pounds per square inch by the least thickness in inches and by the efficiency of the longitudinal riveted joint, and divide by the inside radius of the shell in inches. The result is the pressure per square inch at which the shell should split longitudinally.*

The safe working pressure is found by dividing the above by the desired "factor of safety," usually from 3.5 to 5.

This, it must be noted, is the pressure at which the shell should fail in the manner described. The boiler may be weaker somewhere else, as upon some of the stayed surfaces, so that all these points should be considered before the allowable pressure is fixed upon.

V

SIMPLE TALKS ON THE BRACING OF HORIZONTAL RETURN TUBULAR BOILERS

In former chapters we have discussed the strength of a boiler so far as the parting of the shell is concerned, but even if the shell is heavy enough and the joint well proportioned the boiler may be weak in other respects.

The head of a 60-inch boiler has an area of

$$60 \times 60 \times 0.7854 = 2827 \text{ square inches.}$$

At 100 pounds per square inch there would be a pressure against the head of

$$2827 \times 100 = 282700 \text{ pounds.}$$

or over 140 tons.

Besides its tendency to pull the shell apart endwise, this pressure tends to bulge the heads, as shown in Fig. 22. In the case of a tank, or of the drums of water-tube boilers where there are no tubes in the heads, they can be made safe against change of shape under pressure by giving them in the first place the shape that the pressure tends to force them into; but the tube sheet of a horizontal tubular boiler, for in-

stance, must be flat to allow the tubes to enter square with its surface. The tubes themselves act as stays to the lower part, but the pressure on the part above the tubes tends to bulge the head and might cause the upper tubes to pull out.

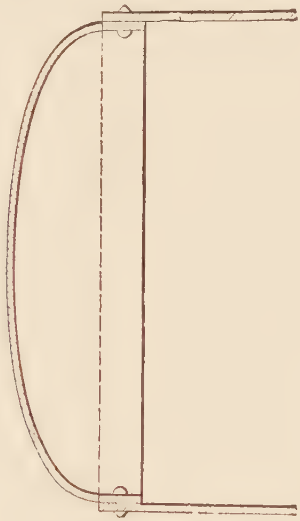

FIG. 22.

This is prevented by bracing the unsupported part of the head either by "through braces," as in Fig. 23, or by "diagonal braces," as in Fig. 24.

In order to find how many braces are required, or if a given boiler is sufficiently braced, the area to be braced must be computed. This area may be taken as that included within lines drawn 2 inches above the top line of tubes and 2 inches inside of the shell,

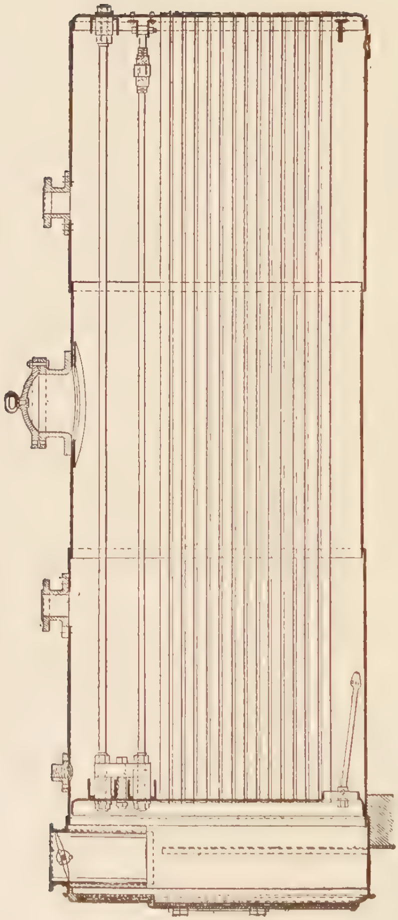

FIG. 23.

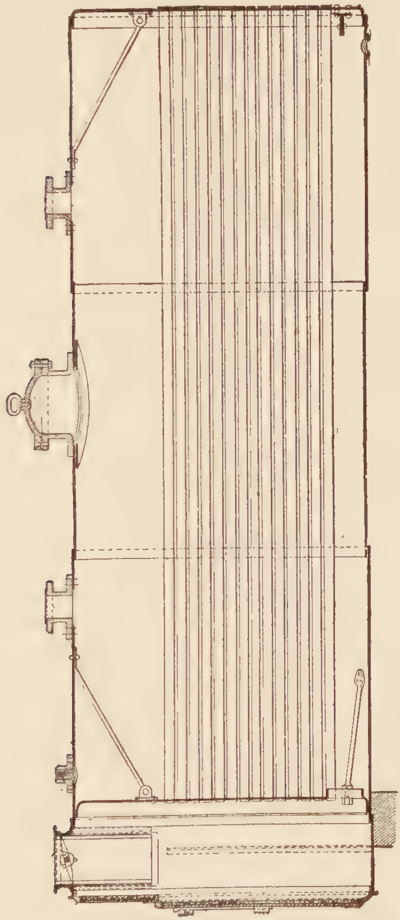

Fig. 24.

as in Fig. 25, the area outside of these lines being considered to be sufficiently braced by the shell and tubes. This figure is a "segment" of a circle and its area is found by dividing its height h by the diameter of the circle, of which it is a part, finding the quotient in the column of "versed sines" of the accompanying table, and multiplying the segmental area as given opposite that quotient in the next column by the square of the diameter.

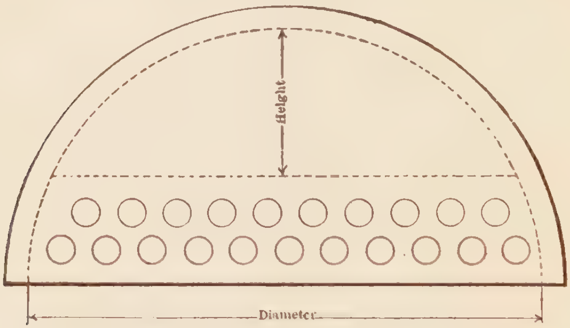

FIG. 25.

For example, suppose the hight h in Fig. 25 to be 18 inches and the diameter of the boiler 60 inches. The diameter of the circle of which the segment is a part is 56 inches, because we go 2 inches inside the shell on both ends of the diameter.

Following the rule we divide the height by the diameter

$$18 \div 56 = 0.3214.$$

The values in the table are given to only three places of decimals, but the division should be carried out to

four places. If the last figure is less than 5, drop it off. If it is 5 and the quotient comes out even, *i.e.*, there is no remainder after the 5, drop it off, also. If the last figure is greater than 5, or in case it is 5, and the division did not come out square, drop it off, but raise the third figure one.

In the case in hand the quotient, 0.3214, comes between the 0.321 and 0.322 of the table, and is nearer the 0.321, being but $0.3214 - 0.321 = 0.0004$ off; while it is $0.322 - 0.3214 = 0.0006$ off from the higher value. If it were 0.3216, however, it would be nearer 0.322 than 0.321.

The segmental area corresponding with 0.321 in the table is 0.2176. Multiplying this by the square of the diameter gives $56 \times 56 \times 0.2176 = 682.39$ square inches as the area of the segment, and the force to be braced against is this number of square inches multiplied by the pressure per square inch.

A through-brace which pulls squarely on the plate has an effect in keeping it from bulging equal to the tensile strain in the brace itself — *i.e.*, if the brace were under a strain of 6000 pounds, it would tend to pull the head in and keep it from bulging with an equal force; but if a diagonal brace, as in Fig. 26, were under a strain of 6000 pounds, it would tend to pull the lug on the head in its own direction with that force, but would resist a force in a direction at right angles with the head of only .91 as much, or 5.460 pounds. This figure is found by dividing the length of the line *b c* by the length of the line *a b*.

In order to find if a boiler is sufficiently braced:

Find the smallest cross-section of each brace in square inches.

Multiply the cross-section of each diagonal brace by the quotient of the distance of its far end from the head in a line perpendicular to the head (*b c*, Fig. 26) divided by the length of the brace.

Add all these results together.

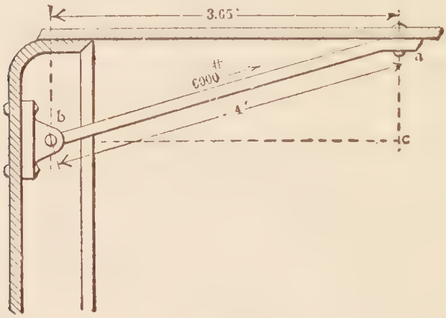

FIG. 26.

For such braces as are all alike, as for through-braces of the same diameter of cross-section, you can of course compute one and multiply it by the number of similar ones.

Divide the product of the area to be braced and the pressure per square inch by the sum of all these values, and you will have the strain on the braces per square inch of section.

The rules of the United States Board of Supervising Inspectors allow a strain of 6000 pounds per square inch on the braces. If the computed stress does not exceed this amount, the boiler is sufficiently braced.

To determine what pressure a boiler will stand, so far as its bracing is concerned, multiply the minimum cross-section of each brace by the quotient of the distance of its far end from the plate perpendicularly divided by the length of the brace. Add the results and multiply by 6000. Divide the produce by the number of square inches in the segment, and the quotient will be the pressure per square inch that the bracing is good for.

AREAS OF SEGMENTS OF CIRCLES

Versed Sine.	Segmental Area.	Versed Sine.	Segmental Area.	Versed Sine.	Segmental Area.	Versed Sine.	Segmental Area.
.1	.04087	.121	.05404	.142	.06892	.163	.08332
.101	.04148	.122	.05469	.143	.06892	.164	.08406
.102	.04208	.123	.05534	.144	.06962	.165	.0848
.103	.04269	.124	.056	.145	.07033	.166	.08554
.104	.0431	.125	.05666	.146	.07103	.167	.08629
.105	.04391	.126	.05733	.147	.07174	.168	.08704
.106	.04452	.127	.05799	.148	.07245	.169	.08779
.107	.04514	.128	.05866	.149	.07316	.17	.08853
.108	.04575	.129	.05933	.15	.07387	.171	.08929
.109	.04638	.13	.06	.151	.07459	.172	.09004
.11	.047	.131	.06067	.152	.07531	.173	.0908
.111	.04763	.132	.06135	.153	.07603	.174	.09155
.112	.04826	.133	.06203	.154	.07675	.175	.09231
.113	.04889	.134	.06271	.155	.07747	.176	.09307
.114	.04953	.135	.06339	.156	.0782	.177	.09384
.115	.05016	.136	.06407	.157	.07892	.178	.0946
.116	.0508	.137	.06476	.158	.07965	.179	.09537
.117	.05145	.138	.06545	.159	.08038	.18	.09613
.118	.05209	.139	.06614	.16	.08111	.181	.0969
.119	.05274	.14	.06683	.161	.08185	.182	.09767
.12	.05338	.141	.06753	.162	.08258	.183	.09845

AREAS OF SEGMENTS OF CIRCLES—*Continued*

Versed Sine.	Segmental Area.	Versed Sine.	Segmental Area.	Versed Sine.	Segmental Area.	Versed Sine.	Segmental Area.
.184	.09922	.216	.12481	.248	.15182	.28	.18002
.185	.1	.217	.12563	.249	.15268	.281	.18092
.186	.10077	.218	.12646	.25	.15355	.282	.18182
187	.10155	.219	.12728	.251	.15441	.283	.18272
188	.10233	.22	.12811	.252	.15528	.284	.18361
.189	.10312	.221	.12894	.253	.15615	.285	.18452
.19	.1039	.222	.12977	.254	.15702	.286	.18542
.191	.10468	.223	.1306	.255	.15789	.287	.18633
192	.10547	.224	.13144	.256	.15876	.288	.18723
.193	.10626	.225	.13227	.257	.15964	.289	.18814
.194	.10705	.226	.13311	.258	.16051	.29	.18905
.195	.10784	.227	.13394	.259	.16139	.291	.18995
.196	.10864	.228	.13478	.26	.16226	.292	.19086
.197	.10943	.229	.13562	.261	.16314	.293	.19177
.198	.11023	.23	.13646	.262	.16402	.294	.19268
.199	.11102	.231	.13731	.263	.1649	.295	.1936
.2	.11182	.232	.13815	.264	.16578	.296	.19451
.201	.11262	.233	.139	.265	.16666	.297	.19542
.202	.11343	.234	.13984	.266	.16755	.298	.19634
.203	.11423	.235	.14069	.267	.16844	.299	.19725
.204	.11503	.236	.14154	.268	.16931	.3	.19817
.205	.11584	.237	.14239	.269	.1702	.301	.19908
.206	.11665	.238	.14324	.27	.17109	.302	.2
.207	.11746	.239	.14409	.271	.17197	.303	.20092
.208	.11827	.24	.14494	.272	.17287	.304	.20184
.209	.11908	.241	.1458	.273	.17376	.305	.20276
.21	.1199	.242	.14665	.274	.17465	.306	.20368
.211	.12071	.243	.14751	.275	.17554	.307	.2046
212	.12153	.244	.14837	.276	.17643	.308	.20553
213	.12235	.245	.14923	.277	.17733	.309	.20645
.214	.12317	.246	.15009	.278	.17822	.31	.20738
.215	.12399	.247	.15095	.279	.17912	.311	.2083

AREAS OF SEGMENTS OF CIRCLES — *Continued*

Versed Sine.	Segmental Area.	Versed Sine.	Segmental Area.	Versed Sine.	Segmental Area.	Versed Sine.	Segmental Area.
.312	.20923	.343	.23832	.374	.26804	.405	.29827
.313	.21015	.344	.23927	.375	.26901	.406	.29925
.314	.21108	.345	.24022	.376	.26998	.407	.30024
.315	.21201	.346	.24117	.377	.27095	.408	.30122
.316	.21294	.347	.24212	.378	.27192	.409	.3022
.317	.21387	.348	.24307	.379	.27289	.41	.30319
.318	.2148	.349	.24403	.38	.27386	.411	.30417
.319	.21573	.35	.24498	.381	.27483	.412	.30515
.32	.21667	.351	.24593	.382	.27580	.413	.30614
.321	.2176	.352	.24689	.383	.27677	.414	.30712
.322	.21853	.353	.24784	.384	.27775	.415	.30811
.323	.21947	.354	.2488	.385	.27872	.416	.30909
.324	.2204	.355	.24976	.386	.27969	.417	.31008
.325	.22134	.356	.25071	.387	.28067	.418	.31107
.326	.22228	.357	.25167	.388	.28164	.419	.31205
.327	.22321	.358	.25263	.389	.28262	.42	.31304
.328	.22415	.359	.25359	.39	.28359	.421	.31403
.329	.22509	.36	.25455	.391	.28457	.422	.31502
.33	.22603	.361	.25551	.392	.28554	.423	.316
.331	.22697	.362	.25647	.393	.28652	.424	.31699
.332	.22791	.363	.25743	.394	.2875	.425	.31798
.333	.22886	.364	.25839	.395	.28848	.426	.31897
.334	.2298	.365	.25936	.396	.28945	.427	.31996
.335	.23074	.366	.26032	.397	.29043	.428	.32095
.336	.23169	.367	.26128	.398	.29141	.429	.32194
.337	.23263	.368	.26225	.399	.29239	.43	.32293
.338	.23358	.369	.26321	.4	.29337	.431	.32391
.339	.23453	.37	.26418	.401	.29435	.432	.3249
.34	.23547	.371	.26514	.402	.29533	.433	.3259
.341	.23642	.372	.26611	.403	.29631	.434	.32689
.342	.23737	.373	.26708	.404	.29729	.435	.32788

VI

CALCULATING THE STRENGTH OF RIVETED JOINTS[1]

In calculations relative to the strength of steam boilers and vessels of a similar character for withstanding high pressures, one of the most important points to be considered is the strength of the seams where the plates are joined. This is not only important to the designer of such vessels, but also to the operating engineer, who is often required to fix the limit of pressure which should be carried on the boilers under his charge, and frequently, owing to increased output without corresponding addition to the boiler capacity, it becomes necessary to carry the pressure as high as safety will permit, and in such cases it is important for the engineer to be able to fix this safe limit.

It is the purpose in this chapter to show how the strength of the various types of joint generally used in boiler construction may be calculated, and as only simple arithmetic is required for the calculations, any reader should find no difficulty in understanding how it is done, and applying the principles to calculate the strength of the particular joints which may be of interest to them. To avoid the use of formulas, which

[1] Contributed to Power by S. F. Jeter.

are confusing to many, numerical examples will be used to illustrate the methods of making the calculations, and for the sake of uniformity the tensile strength of the sheets (which is the strength to resist being pulled apart) will be assumed as 55,000 pounds per square inch; the shearing strength of the rivets (which represents their resistance to being sheared through by the plates at right angles to their length) will be assumed as 42,000 pounds per square inch in single shear, as represented in Fig. 31, and 78,000 pounds per square inch in double shear, as represented in Fig. 34. The resistance of the rivets to crushing will be assumed at 95,000 pounds per square inch. For modern construction consisting of steel plates and steel rivets, the above values are average figures.

It is customary to express the strength of a riveted joint as a percentage of the strength of the plates which are riveted together. Thus, if the joint illustrated in Fig. 35 has an efficiency of $62\frac{1}{2}$ per cent., it would mean that any portion of its length that divides the rivet spaces symmetrically would be 0.625 times as strong as a section of the same length through the solid plate.

Possible Modes of Failure

Before proceeding to calculate a practical boiler joint, the different ways in which two pieces of plate riveted together might fail should be noted. If a piece of boiler plate, $\frac{3}{8}$ inch thick and $2\frac{1}{2}$ inches wide, is placed in the jaws of a testing machine, as illustrated in Fig. 27, and pulled apart, it would separate at some section as *A A*. If the tensile strength was 55,000 pounds per

square inch, the force that would have to be applied to the jaws would be 55,000 times the area separated in square inches, which in this case is

$$2\tfrac{1}{2} \times \tfrac{3}{8} = \tfrac{15}{16} = 0.9375$$

square inch, so that the pull would be

$$55{,}000 \times 0.9375 = 51{,}562.5$$

pounds.

If another piece of plate be taken, identical in every

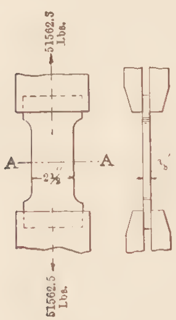

FIG. 27.

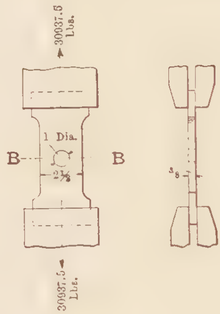

FIG. 28.

respect to the first, except that a hole 1 inch in diameter is drilled through it as illustrated in Fig. 28, and the plate be pulled apart in the testing machine as before, it is evident that it would fail along the line *B B*, as the area of the reduced section caused by drilling the hole would be only

$$(2.5 - 1) \times 0.375 = 0.5625$$

square inch, and the force necessary to pull it apart would be

$$55{,}000 \times 0.5625 = 30{,}937.5$$

pounds, the strength of the metal being the same in both instances. Now if the relation between the strength of the solid plate and the drilled plate be expressed by dividing the latter by the former, the result would be

$$\frac{30,937.5}{51,562.5} = 0.6,$$

or, in other words, the drilled plate is capable of sustaining 60 per cent. of the load that could be carried by the solid plate.

If, instead of using a single piece of plate, two plates are drilled with 1-inch holes in the ends and are joined together by a rivet, as shown in Fig. 29, and an attempt

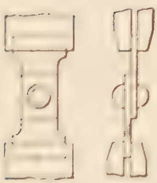

FIG. 29.

FIG. 30.

should be made to pull them apart as before, there would be four probable ways in which failure might take place, all of which are considered in the calculation and design of riveted joints. First, the section of plate each side of the rivet hole might break, leaving the ends

as shown in Fig. 30. Again, the plates might shear the rivets off, as illustrated in Fig. 31. Thirdly, it has been found by practical tests of joints that steel rivets cannot be subjected to a pressure much greater than 95,000 pounds per square inch of bearing surface without materially affecting their power to resist shearing, and therefore the joint might fail, as shown in Fig. 31, due to an excess crushing stress on the rivet.

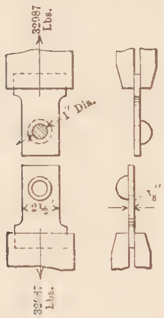

Fig. 31.

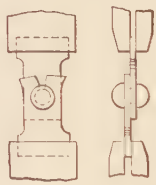
Fig. 32.

A fourth possible method of failure would be for the metal in the sheet in front of the rivet to split apart or pull out, as illustrated in Fig. 32. This latter mode of failure is erratic, and cannot be calculated, but it has been practically demonstrated in tests of joints, that if the distance from the edge of the plate to the center of the rivet hole is $1\frac{1}{2}$ times the diameter of the hole, this mode of failure is improbable, and in the following calculations of joints it will be assumed that they are properly designed to render such failure impossible.

To determine the actual strength or efficiency of such a joint as is illustrated in Fig. 29, the force required to produce rupture must be calculated for each of the first three ways mentioned, and the weakest mode of failure taken as the maximum strength of the joint.

To rupture the plate as illustrated in Fig. 30, the pull required would be the same as to rupture the drilled plate illustrated in Fig. 28, which was found to be 30,937.5 pounds. To shear the rivet off, as in Fig. 31, would require a force equal to the area to be sheared in square inches, times the shearing strength per square inch; or since the area of a 1-inch rivet is 0.7854 square inch, the force required would be

$$0.7854 \times 42{,}000 = 32{,}987$$

pounds. The pressure required to cause failure by crushing was stated to be 95,000 pounds per square inch, and in calculating the area exposed to pressure for pins and rivets, it is figured as equal to the diameter of the pin or rivet, times the thickness of the plate; therefore, we have

$$1 \times 0.375 = 0.375$$

square inch of area to withstand crushing, or

$$0.375 \times 95{,}000 = 35{,}625$$

pounds would be required to produce rupture of the joint in this manner.

From these figures it is evident that the method of failure first considered is the weakest of the three

and, therefore, determines the efficiency of the joint, which would be 60 per cent. as found for the drilled plate.

If one plate is riveted between two other plates, as illustrated in Fig. 33, the several methods of failure are calculated in the same way, except for the shearing of the rivet, which would occur as shown in Fig. 34, and

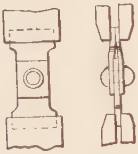

FIG. 33.

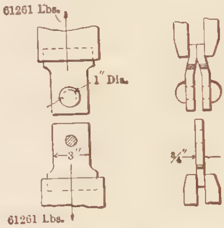

FIG. 34.

is described as double shearing. While the metal sheared in this case would be just twice as much as in single shear, it has been found by test that the force required is not exactly twice as much, but 1.85 to 1.90 times the amount in single shear; so as stated at the beginning, 78,000 pounds per square inch is assumed for rivets in double shear and 42,000 pounds per square inch when in single shear.

Calculating the strength of a joint with the dimensions as illustrated in Fig. 34, the strength of the solid plate would be

$$3 \times 0.75 \times 55{,}000 = 123{,}750$$

pounds. The strength of the center plate through the

rivet hole, the failure being assumed similar to that illustrated in Fig. 30, would be

$$(3 - 1) \times 0.75 \times 55{,}000 = 82{,}500$$

pounds. The crushing strength of the rivet would be

$$1 \times 0.75 \times 95{,}000 = 71{,}250$$

pounds. The shearing strength of the rivet, or failure assumed as in Fig. 34, would be

$$0.7854 \times 78{,}000 = 61{,}261$$

pounds.

From these figures it is evident that failure would most likely occur as shown in Fig. 34, and the relative strength of the joint as compared with the solid plate is

$$\frac{61{,}261}{123{,}750} = 0.495,$$

or $49\frac{1}{2}$ per cent. As will be shown later, the foregoing simple calculations are all that are required to estimate the strength of the most complicated joints.

The Unit Section

In calculating the strength of a practical boiler joint, the strength for the entire length of a sheet could be estimated, but this would be laborious owing to the number of figures involved in the calculations, and the same result can be obtained by considering any length that divides the rivets symmetrically. For convenience, the shortest length that thus divides the rivets is the one used in such calculations, and this length is called a unit section of the joint. When the lines dividing the

joint into unit sections pass through a rivet, only one-half of the rivet is considered in the calculation, and when rivets thus divided are lettered for reference, the two halves on opposite sides will be lettered the same, so that referring to the letter will indicate a whole rivet. Thus, if the rivet *A*, in Fig. 43, is spoken of, it would mean the combined halves of the two rivets on the outer row.

In measuring joints already constructed to obtain the length of a unit section, or the pitch, it should be remembered that rivet heads do not always drive fairly over the center of the rivet holes, and the rivet holes themselves are sometimes irregular distances apart; so it is more accurate to measure a number of pitches and divide the distance by the number measured to obtain the average pitch. It will be found most convenient, where space permits, to measure ten pitches, and then placing the decimal point one figure to the left will give the average unit length.

Single-riveted Lap-joint

First to be considered is the single-riveted lap-joint illustrated in Fig. 35. In a unit section of 2 inches one rivet is in single shear and ¾ inch has been cut out of the plate by the rivet hole. The calculation for strength is the same as has been made for Fig. 38, and the three methods of failure to be considered are:

(1) Breaking of the section of plate between the rivet holes, which is called the net section.

(2) Shearing of a ¾-inch rivet in single shear.

(3) Resistance of one rivet to crushing.

Using the numerical values given in Fig. 35 the following results are obtained:

(1) $(2 - 0.75) \times 0.25 \times 55{,}000 = 17{,}187.5$ pounds.
(2) $0.4418 \times 42{,}000 = 18{,}556$ pounds.
(3) $0.75 \times 0.25 \times 95{,}000 = 17{,}812.5$ pounds.

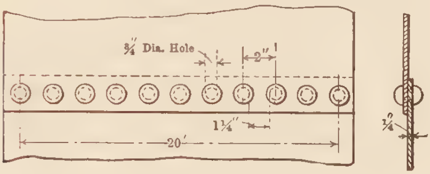

FIG. 35.

Of the three methods of failure, the first is seen to be the most probable, and since a unit section length of the solid plate would have a strength of

$$2 \times 0.25 \times 55{,}000 = 27{,}500$$

pounds, the efficiency of the joint would be

$$\frac{17{,}187.5}{27{,}500} = 62.5$$

per cent.

DOUBLE-RIVETED LAP-JOINT

Next in line is the double-riveted lap-joint illustrated in Fig. 36. There is one feature connected with this joint which should be considered before proceeding with the calculation of its strength. It would evidently be possible to have the two rows of rivets forming this joint so close together that the combined net

sections between rivets $A\ B$ and $B\ C$ would be less than between rivets $A\ C$. It has been found in practical tests of joints that it is necessary to have the combined area of these two sections 30 to 35 per cent. in excess of that between rivets A and C in order to be sure that the joint will fail along line $A\ C$. This would

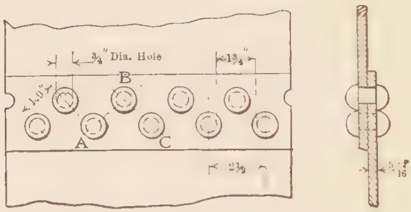

FIG. 36.

correspond to a diagonal pitch of two-thirds of the pitch from A to C plus one-third of the diameter of the rivet hole, or 1.9 inches in the joint shown in Fig. 36. Ordinarily, if the rows are much closer than this, the joint has an abnormal appearance which would be noted at once. In further calculations it will be assumed that the joints are proportioned so that this method of failure will not be possible.

Proceeding with the calculation of the strength of the joint illustrated in Fig. 36, the methods of probable failure to be calculated are the same as for the single-riveted joint:

(1) Failure of net section between rivet holes.
(2) Shearing of two rivets in single shear.
(3) Crushing strain on two rivets.

Using the values given in Fig. 36, we have for the above:

(1) $(2.5 - 0.75) \times 0.3125 \times 55{,}000 = 30{,}078$ pounds.

(2) $2 \times 0.4418 \times 42{,}000 = 37{,}112$ pounds.

(3) $2 \times 0.75 \times 0.3125 \times 95{,}000 = 44{,}531$ pounds.

It is evident that the first method of failure is the most probable, and since the strength of the solid plate is

$$2.5 \times 0.3125 \times 55{,}000 = 42{,}969$$

pounds, the efficiency of the joint will be

$$\frac{30{,}078}{42{,}969} = 70$$

per cent.

Triple-riveted Lap-joint

In Fig. 37 is illustrated a triple-riveted lap-joint. Here the length of unit section is 3 inches, and the different probable modes of failure are identical with those of the single- and double-riveted lap-joints except in rivet strength. It will be noted that in this case there are three rivets contained in each unit section, which are subjected to shear and crushing. The several methods of probable failure to be investigated are as follows:

(1) Failure of net section between the rivet holes of outer rows.

(2) Shearing of three rivets in single shear.

(3) Crushing strain on three rivets.

Using the numerical values specified in Fig. 37, we would have:

(1) $(3 - 0.75) \times 0.375 \times 55{,}000 = 46{,}406$ pounds.
(2) $3 \times 0.4418 \times 42{,}000 = 55{,}667$ pounds.
(3) $3 \times 0.75 \times 0.375 \times 95{,}000 = 80{,}156$ pounds.

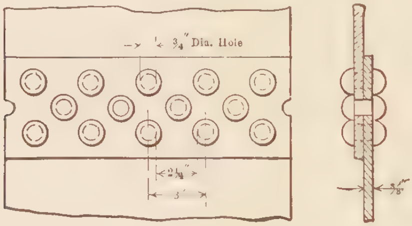

FIG. 37.

The first method of failure assumed is the most likely, and as the strength of the solid plate for a unit section of length is

$$3 \times 0.375 \times 55{,}000 = 61{,}875$$

pounds, the efficiency of joint is

$$\frac{46{,}406}{61{,}875} = 75$$

per cent. The triple-riveted joint represents about the maximum strength that can be obtained in practice from simple lap-riveted joints, as in this form the maximum pitch distance that permits proper calking of the edge of the plates is reached, and still leaving

the net section of metal between the rivet holes the weakest portion of the joint, so that further addition of rivets would not add to its strength.

Chain Riveting

Joints illustrated in Figs. 36 and 37 have the rivets arranged so that the rivets in one row come opposite the

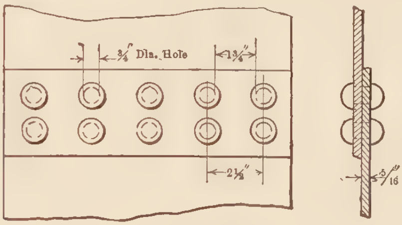

Fig. 38.

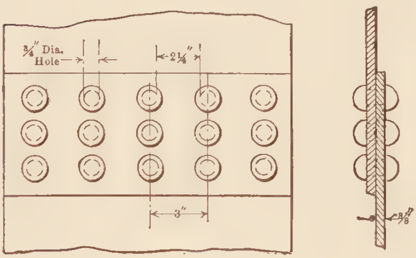

Fig. 39.

spaces in the adjacent rows, and this arrangement is termed staggered riveting. The same forms of joint are sometimes made with the rivets placed in straight rows across the joint, is illustrated in Figs. 38 and 39, which is known as chain riveting. The calculations for joint

efficiency in chain-riveted joints are identical in every respect to those for staggered riveting, and with equal diameters and spacing of rivets and equal thicknesses of plate, the efficiencies are the same for either type.

Lap-riveted Joint with Inside Strap

While the lap-riveted joint with inside strap is not extensively used in the manufacture of new boilers, it affords a ready means of strengthening simple lap

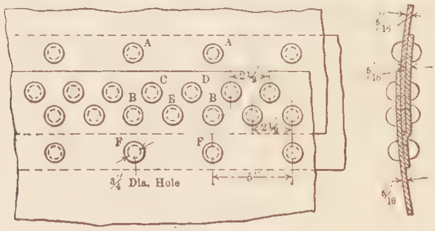

Fig. 40.

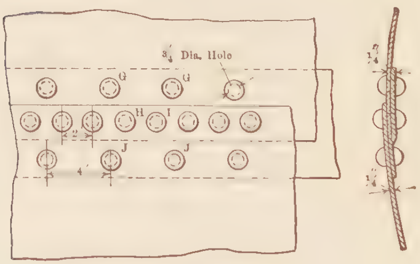

Fig. 41.

seams on boilers already constructed, and it is quite extensively used for this purpose. This joint is illustrated in Figs. 40 and 41, and it will be seen that it is equally applicable to single- or double-riveted

lap-joints; it could also be applied to triple-riveted joints. The joint illustrated in Fig. 40 is identical in every respect with the one shown in Fig. 36, excepting the addition of the $\frac{5}{16}$-inch cover strip and the outer rows of rivets, these dimensions being selected to facilitate comparison between the strengths of the two joints. A unit section of this joint is 5 inches long, and five methods of failure present themselves for consideration in determining the strength of the joint:

(1) Breaking of the plate along the section between the rivet holes *A A*.

(2) The separation of the plate along the section on line of rivets *C D* and shearing the rivet *A*.

(3) Separation of the plate along the section on line of rivets *C D* and the crushing of rivet *A*.

(4) Crushing of rivets *A B C D E* by the shell.

(5) Shearing of rivets *B C D E F* in single shear.

The pulling out of the upper plate, which would shear rivet *A* single, and *B C D E* double, need not be considered, since it would evidently be stronger than the first method considered above. Calculating the value of the possible methods of failure by using the dimensions given in Fig. 40, we have:

(1) $(5 - 0.75) \times 0.3125 \times 55{,}000 = 73{,}040$ pounds.

(2) $[5 - (2 \times 0.75)] \times 0.3125 \times 55{,}000 + 0.4418 \times 42{,}000 = 78{,}726$ pounds.

(3) $[5 - (2 \times 0.75)] \times 0.3125 \times 55{,}000 + 0.3125 \times 0.75 \times 95{,}000 = 82{,}436$ pounds.

(4) $0.3125 \times 0.75 \times 95{,}000 \times 5 = 111{,}330$ pounds.

(5) $0.4418 \times 42{,}000 \times 5 = 92{,}780$ pounds.

Evidently the next section between the rivet holes *A A* is the weakest portion of the joint, and since a section of the solid plate 5 inches long has a strength of

$$5 \times 0.3125 \times 55{,}000 = 85{,}937$$

pounds, the efficiency of the joint is

$$\frac{73{,}040}{85{,}937} = 85$$

per cent.

Calculation of the joint illustrated in Fig. 41 is proceeded with in the same manner as for Fig. 40. It will be noted that to aid comparison the dimensions have been assumed the same as in Fig. 35 with the strap added. The methods of possible failure to be compared are:

(1) Separation of the plate along net section *G G*.

(2) Separation of plate along section *H I* and shearing of rivet *G* in single shear.

(3) Separation of plate along section *H I* and crushing of rivet *G*.

(4) Crushing of rivets *G H I*.

(5) Shearing of rivets *H I J* in single shear.

According to the dimensions given in Fig. 41 the numerical values would give the following results.

(1) $(4 - 0.75) \times 0.25 \times 55{,}000 = 44{,}687$ pounds.

(2) $[4 - (2 \times 0.75)] \times 0.25 \times 55{,}000 + 0.4418 \times 42{,}000 = 52{,}931$ pounds.

(3) $[4 - (2 \times 0.75)] \times 0.25 \times 55{,}000 + 0.75 \times 0.25 \times 95{,}000 = 51{,}187$ pounds.

(4) $0.75 \times 0.25 \times 95{,}000 \times 3 = 53{,}436$ pounds.

(5) $0.4418 \times 42{,}000 \times 3 = 55{,}668$ pounds.

Since the strength of the solid plate is

$$4 \times 0.25 \times 55{,}000 = 55{,}000$$

pounds, the efficiency would be

$$\frac{44{,}687}{55{,}000} = 81.25$$

per cent.

It is thus apparent that by adding a strap to the joint illustrated in Fig. 35 and making it like Fig. 41, the efficiency has been increased from 62.5 per cent. to 81.25 per cent., which would permit an increase in steam pressure of 30 per cent. on the boiler after such change.

Single-riveted Double-strapped Butt-joint

In describing all forms of butt-joints it is customary to refer to the rivets on one side of the butt only; thus, in Fig. 42 there are actually two rows of rivets, but

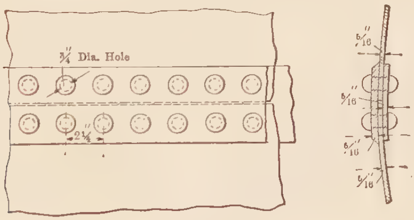

Fig. 42.

the joint is only single-riveted, for the strength of the joint along either row is in no wise dependent on the other row. If the two rows should not be riveted alike, it would be necessary to consider each side as a separate joint to find which was the weaker, in order to deter

mine the strength of the combination. This, however, is not necessary in practical boiler joints, since they are constructed alike on each side of the butt.

In the joint illustrated in Fig. 42 it will be noted that all of the rivets are in double shear, and only three methods of possible failure are presented for calculation:

(1) Breaking the net section.
(2) Shearing of one rivet in double shear.
(3) Crushing of a rivet by the shell.

With the dimensions given in the figure we have:

(1) $(2.25 - 0.75) \times 0.3125 \times 55{,}000 = 25{,}781$ pounds.

(2) $0.4418 \times 78{,}000 = 34{,}460$ pounds.

(3) $0.75 \times 0.3125 \times 95{,}000 = 22{,}230$ pounds.

The strength of the solid plate is

$$2.25 \times 0.3125 \times 55{,}000 = 38{,}672$$

pounds, and since the weakest portion of the joint is the resistance to crushing of the rivets, the efficiency is

$$\frac{22{,}230}{38{,}672} = 57.5$$

per cent.

Double-riveted Double-strapped Butt-joint

Double-riveted butt-joints can be made in two forms, the one generally used being illustrated in Fig. 43. The calculations for the efficiency of this joint are the same as for the single-riveted joint, except that there

are two rivets to be considered in each unit section of the joint instead of one. The three methods of possible failure are:

(1) Pulling apart of the sheet along net section *A A*.
(2) Shearing of rivets, *A B*, in double shear.
(3) Crushing of rivets *A B*.

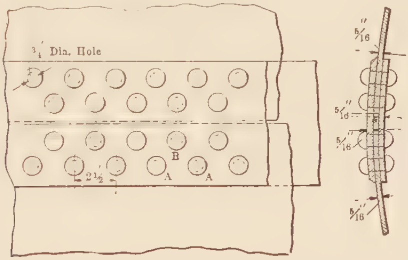

FIG. 43.

Substituting the values given in Fig. 17, we have:

(1) $(2.5 - 0.75) \times 0.3125 \times 55{,}000 = 29{,}085$ pounds.
(2) $0.4418 \times 78{,}000 \times 2 = 68{,}920$ pounds.
(3) $0.75 \times 0.3125 \times 95{,}000 \times 2 = 44{,}532$ pounds.

The strength of the solid plate is

$$2.5 \times 0.3125 \times 55{,}000 = 42{,}969$$

pounds, and the weakest portion of the joint is the net section between rivets *A A*. Therefore, the efficiency is

$$\frac{29{,}085}{42{,}969} = 67.6$$

per cent.

In Fig. 44 is illustrated the second type of double-riveted butt-joint. This form of joint, if proportioned properly, can be made considerably stronger than the one illustrated in Fig. 43. There are six methods of possible failure to be considered:

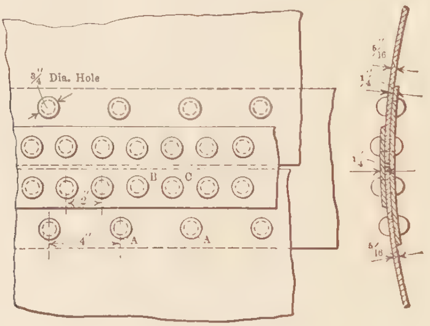

FIG. 44.

(1) Pulling apart of the sheet along net section *A A*.

(2) Pulling apart of the sheet along section *B C* and shearing rivet *A*.

(3) Pulling apart of sheet along section *B C* and crushing of rivet *A*. (Note that in calculating the crushing of rivet *A* the thickness of the strap is to be used instead of the plate, owing to the strap being thinner than the plate.)

(4) Shearing of rivet *A* single and *B*, *C* double shear.

(5) Crushing of rivets *B C* in the plate and *A* in the strap.

(6) Crushing of rivets *B C* in the plate and shearing of rivet *A*.

Substituting the numerical values from Fig. 44, we have:

(1) $(4 - 0.75) \times 0.3125 \times 55{,}000 = 55{,}859$ pounds.

(2) $[(4 - 1.5) \times 0.3125 \times 55{,}000] + (42{,}000 \times 0.4418) = 61{,}525$ pounds.

(3) $[(4 - 1.5) \times 0.3125 \times 55{,}000] + (0.75 \times 0.25 \times 95{,}000) = 60{,}781$ pounds.

(4) $(42{,}000 \times 0.4418) + (78{,}000 \times 0.4418 \times 2) = 87{,}476$ pounds.

(5) $(0.75 \times 0.3125 \times 95{,}000 \times 2) + (0.75 \times 0.25 \times 95{,}000, = 62{,}272$ pounds.

(6) $(0.75 \times 0.3125 \times 95{,}000 \times 2) + (42{,}000 \times 0.4418) = 63{,}087.25$ pounds.

From these figures it will be seen that the net section between the rivet holes *A A* is the one most likely to fail, and since the strength of a unit section of the solid plate is

$$4 \times 0.3125 \times 55.000 = 68{,}750$$

ponuds, the efficiency of the joint is

$$\frac{55{,}859}{68{,}750} = 81.25$$

per cent.

Triple-riveted Double-strapped Butt-joint

The joint illustrated in Fig. 45 is known as the triple-riveted butt-joint. The methods of failure to be investigated are the same as those in Fig. 44, and are as follows:

(1) Pulling apart of sheet at net section *A A*.

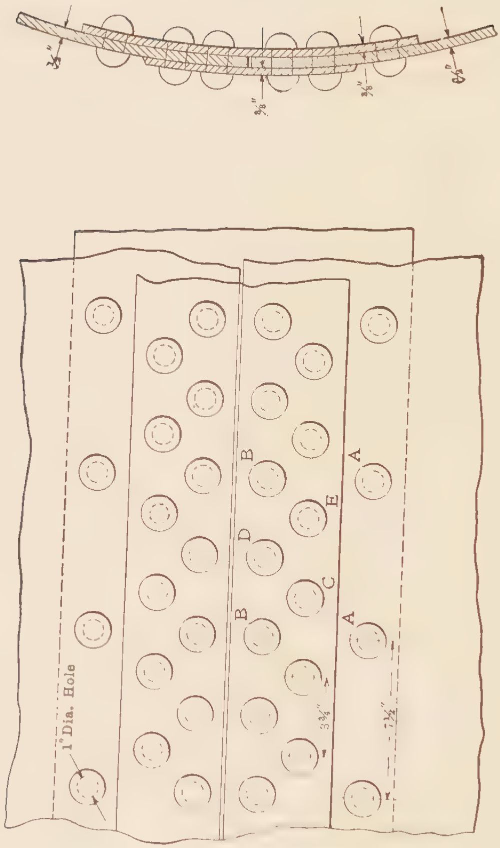

FIG. 45.

(2) Pulling apart of sheet along section *C E* and shearing rivet *A*.

(3) Pulling apart of sheet along section *C E* and crushing rivet *A*.

(4) Shearing rivet *A* single and *B C D E* double.

(5) Crushing of rivets *B C D E* in the plate and *A* in the strap.

(6) Crushing of rivets *B C D E* in the plate and shearing of rivet *A*.

Substituting the values given in Fig. 45:

(1) $(7.5 - 1) \times 0.5 \times 55{,}000 = 178{,}750$ pounds.

(2) $[(7.5 - 2) \times 0.5 \times 55{,}000] + (42{,}000 \times 0.7854) = 184{,}237$ pounds.

(3) $[(7.5 - 2) \times 0.5 \times 55{,}000] + (1 \times 0.5 \times 95{,}000) = 198{,}250$ pounds.

(4) $(0.7854 \times 42{,}000) + (0.7854 \times 78{,}000 \times 4) = 278{,}027$ pounds.

(5) $(1 \times 0.5 \times 95{,}000 \times 4) + (1 \times 0.375 \times 95{,}000) = 225{,}625$ pounds.

(6) $(1 \times 0.5 \times 95{,}000 \times 4) + (0.7854 \times 42{,}000) = 222{,}987$ pounds.

For a unit length the strength of the solid plate is

$$7.5 \times 0.5 \times 55{,}000 = 206{,}250$$

pounds. The net section between rivets *A*, *A* is the weakest portion of the joint, so that the efficiency is

$$\frac{178{,}750}{206{,}250} = 86.7$$

per cent.

Quadruple-riveted Double-strapped Butt-joint

The last type of joint to be considered is the quadruple-riveted butt-joint illustrated in Fig. 46. This joint is now used on nearly all high-grade boilers of the horizontal return-tubular type, and it marks about the practical limit of efficiency for riveted joints connecting plates of uniform thickness together. The methods of failure to be considered are practically the same as in the two preceding joints, except that there are more rivets concerned in the calculations:

(1) Pulling apart of the sheets along net section *A A*.

(2) Pulling apart of the sheet along section *D E F G* and shearing rivets *A B C*.

(3) Pulling apart of sheet along section *D E F G* and crushing of rivets *A B C* in the strap.

(4) Shearing rivets *A B C* in single shear and *D E F G H I J K* in double shear.

(5) Crushing of rivets *D E F G H I J K* in plate and *A B C* in the strap.

(6) Crushing of rivets *D E F G H I J K* in the plate and shearing rivets *A B C*.

Using the numerical values of Fig. 46, we have:

(1) $(15.5 - 1) \times 0.5625 \times 55{,}000 = 448{,}580$ pounds.

(2) $[(15.5 - 4) \times 0.5625 \times 55{,}000] + (3 \times 42{,}000 \times 0.7854) = 454{,}739$ pounds.

(3) $[(15.5 - 4) \times 0.5625 \times 55{,}000] + (3 \times 0.4375 \times 1 \times 95{,}000) = 480{,}465$ pounds.

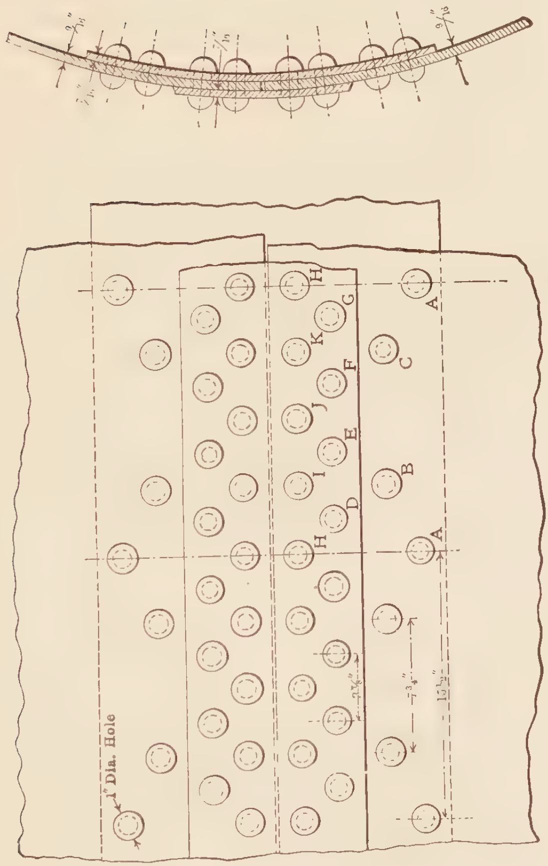

Fig. 46.

(4) $(3 \times 42{,}000 \times 0.7854) + (8 \times 78{,}000 \times 0.7854) = 589{,}050$ pounds.

(5) $(8 \times 0.5625 \times 1 \times 95{,}000) + (3 \times 0.4375 \times 1 \times 95{,}000) = 552{,}187$ pounds.

(6) $(8 \times 0.5625 \times 1 \times 95{,}000) + (3 \times 42{,}000 \times 0.7854 = 526{,}461$ pounds.

The strength of the solid plate is

$$15.5 \times 0.5625 \times 55{,}000 = 479{,}528$$

pounds, and the failure of the sheet by pulling apart along the net section *A A* is the one that determines the efficiency of the joint, which is

$$\frac{448{,}580}{479{,}528} = 93.55$$

per cent.

From the foregoing calculations it may be observed that estimating the efficiency of riveted joints, while very simple, is a rather tedious process, particularly if many joints are to be calculated

VII

TO FIND THE AREA TO BE BRACED IN THE HEADS OF HORIZONTAL TUBULAR BOILERS

FOR the purpose of determining the number of braces to be used, it is not necessary to figure the area of a boiler head to a fraction of a square inch, and a simple rule, the reason for which is so plain that it can never be forgotten, will be helpful to the candidate before the examiner, or when a table of circular segments is not to be had.

The diameter of the boiler and the hight above the top row of tubes are the only measurements which are ordinarily given. The flange is considered good for three inches around the outside, and the tubes for two inches above their top edges, so that the area to be braced is a part of a circle having a diameter six inches less than the given diameter of the boiler and a hight 5 inches less than that of the undiminished segment, which area is represented by the shaded area in Fig. 47.

The area of a circle is the diameter multiplied by itself and by 0.7854. It is easy, then, to find the area of the circle of which the shaded area is a part. Suppose we are dealing with a 72-inch boiler. Allowing for 3 inches on each end of the diameter, the diameter

of the circle of which the segment to be braced is a part would be

$$72 - 6 = 66 \textit{ inches},$$

and its area would be

$$66 \times 66 \times 0.7854 = 3421 \text{ square inches};$$

and the area of the half circle *a b c d e* would be one-half of this, or 1710 square inches.

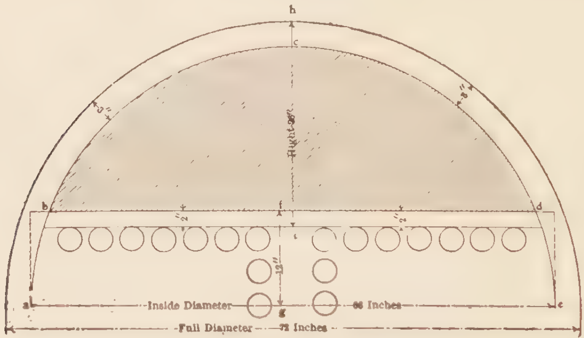

FIG. 47.

Now, if from this area the area *a b d e* is subtracted, the remainder will be the required area of the (shaded) portion to be braced. The hight *f g* is the radius, or one-half the given diameter less the given hight plus 2, and it will be near enough if we consider its length equal to the diameter, as the length of the chord *b d* is not usually given. Suppose the hight *h i* to be 26 inches, then the hight *f g* of the portion to be subtracted would be

$$\frac{72}{2} - 26 + 2 = \text{inches},$$

and if its length be taken at 66 inches its area will be

$$12 \times 66 = 792 \text{ square inches.}$$

This is too great by the area of the two little dotted triangles at *a b* and *d e*, but this is so small a proportion of the total area that it may be neglected, especially if it is borne in mind when deciding upon the number of braces that the area as determined is a little small.

Subtracting this area from that of one-half the 66-inch circle, as above found, we have

$$1710 - 792 = 918 \text{ square inches}$$

as the area to be braced.

If the pressure is 100 pounds per square inch, the force to be braced against is

$$918 \times 100 = 91{,}800 \text{ pounds,}$$

and if the braces used are good for 8000 pounds apiece, it will take

$$91{,}800 \div 8000 = 11.5 \text{ braces.}$$

We should have to use 12 braces, anyway, and these would be good for

$$\frac{12 \times 8000}{100} = 960 \text{ inches,}$$

while the actual area is 936, instead of 918, as the above approximate method made it. Unless the number of braces comes out very nearly square in the calculation, there will be enough leeway in using a whole brace for the fraction to make up for the shortness of the area. When this fraction exceeds, say, 0.9, safety would be insured by putting in an extra brace.

VIII

GRAPHICAL DETERMINATION OF BOILER DIMENSIONS[1]

THE variables entering into the design of a steam boiler shell are the working pressure, the diameter of the shell, the thickness and tensile strength of the plate, the diameter, spacing and shearing value of the rivets, the efficiency of the joints and the factor of safety.

The usual working pressures are 80, 100, 125, and 150 pounds per square inch.

The standard diameters of shell are 44, 48, 54, 60, 66, 72, 78, 84, 90 and 96 inches.

The tensile strength of the plate is 52,000 to 62,000 pounds per square inch for flange steel and 55,000 to 65,000 pounds per square inch for fire-box steel. The average assumed for calculations is 60,000 pounds per square inch. The shearing value of steel rivets is 38,000 to 42,000 pounds per square inch. Until recently 38,000 pounds per square inch was used for all calculations, but this value has been gradually increasing with the improved quality of steel rivets, until 42,000 pounds per square inch is now the more generally accepted value.

[1] Contributed to Power by N. A. Carle.

This has resulted in an increased spacing of rivets, together with an increase in the efficiency of the joints, and a consequent reduction in the thickness of plate.

Rivet holes are usually punched $\frac{1}{16}$-inch larger than the rivets and calculated as $\frac{1}{8}$-inch larger than the rivets.

In marine practice, holes are specified as drilled or punched $\frac{1}{16}$-inch small, the shell assembled and the holes then reamed to full size.

The shearing value of the rivet is calculated for the stock size before driving.

The crushing value of steel rivets has been practically eliminated from the problem, because in practice the sizes selected give values in excess of the shearing value.

No consideration is given to the friction of the joint, it being assumed that this is all destroyed before rupture, so that it is not a factor of the ultimate strength.

The kind of joints and size and spacing of the rivets are governed by accident insurance companies' requirements and shop practice.

The size of rivets and spacing used necessary to insure good calking usually make the horizontal joint the weakest point in the boiler and therefore the governing factor.

It is desirable to get a high efficiency of the joint for high pressures and thick plates. Different types of joints are designated as single lap-riveted, double lap-riveted, triple lap-riveted, double butt-strap-riveted, triple butt-strap-riveted and quadruple butt-strap-riveted.

The single lap-riveted joint is used on girth seams generally, as the stress is only one-half that on the horizontal joint, and on the horizontal seams only for very small diameters and pressures.

The quadruple butt-strap-riveted joint is used only on very heavy plate, large diameters and high pressures.

The efficiencies depend upon the rivet spacing, diameter of rivets and the allowances and assumptions made.

Design conditions reduce the problem to the efficiency of the joint based on tearing between the outer row of rivets.

The usual efficiencies used in calculations in the shell formula are double lap, 70 per cent.; triple lap, 75 per cent.; double butt-strap, 80 per cent., and triple butt-strap, 86 per cent.

The factors of safety ordinarily used are 4, $4\frac{1}{2}$ and 5, with 6 sometimes specified in marine practice.

The shell formula is

$$D. \times W.P. \times F.S. = 2 \times S. \times E. \times t.$$

$D.$ = Diameter of shell in inches.
$W.P.$ = Working pressure in pounds per square inch.
$F.S.$ = Factor of safety.
$E.$ = Efficiency of horizontal joint.
$t.$ = Thickness of plate in inches.

These are shown graphically in the calculating diagram (Fig. 48). The use of this diagram is probably best illustrated by an example:

Given. — The boiler shell 66 inches in diameter

for a working pressure of 125 pounds with a factor of safety of 5. What thickness of plate is required for the shell?

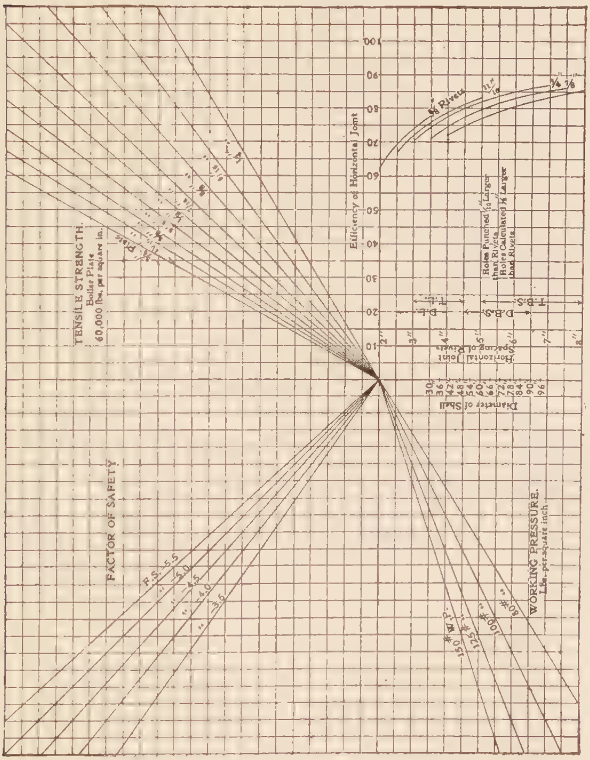

FIG. 48.

Assume that a double butt-strap joint will be used with an efficiency of 80 per cent. Starting with 66 inches "diameter of shell," read across to 125 pounds

"working pressure," then up to a "factor of safety" of 5, and then across to its intersection with a vertical line through 80 per cent. "efficiency of joint." This gives a value slightly less than $\frac{7}{16}$ inch for "thickness of plate."

Hence use $\frac{7}{16}$ inch and by reading back it will be found that this gives about 5.1 as factor of safety.

Usually the designer has shop practice to follow, so that instead of using approximate values for the efficiency, the usual spacing and diameter of rivets can be selected and the actual efficiency obtained. As an example, assume that for a double butt-strap-riveted joint the shop spacing was $4\frac{1}{2}$ inches and $2\frac{1}{4}$ inches, using $\frac{11}{16}$-inch rivets. Read across from $4\frac{1}{2}$ inches "spacing of rivets" to $\frac{11}{16}$-inch rivets and then up to 82 per cent. "efficiency of horizontal joint." The boiler heads are made $\frac{1}{16}$ inch thicker than the shell, as the metal is decreased about this amount in dishing and flanging the head. The spacing of the girth-seam rivets is according to shop practice and does not require a high efficiency, as the stress is only one-half that of the shell. It is, therefore, a dependent factor in the design.

IX

THE SAFETY VALVE

THE study of the safety valve has been the first step of many a man in scientific engineering. Induced to its study by the necessity of solving its problems before the examiner, his consideration of this simple device has led him into the computation of areas, into a study of the principle of the lever, of moments of forces, of the velocity of flow of steam and other fundamental principles of mechanics. This applies to those who have studied the subject intelligently, not to those who have attempted to get over it by learning a rule by rote, simply to be confounded when confronted by another rule, or a case to which their rule would not apply. The whole subject is so simple that an hour's study will put a man in possession of the fundamentals so that he can make his own rules or solve any problem without a rule, from a sheer understanding of the principles involved.

PRESSURE PER SQUARE INCH

A cubic foot of water weighs, in round numbers, 62 pounds. If you can imagine ten cubic feet packed one above the other, as in Fig. 49, they would make a column weighing some 620 points, supported on a

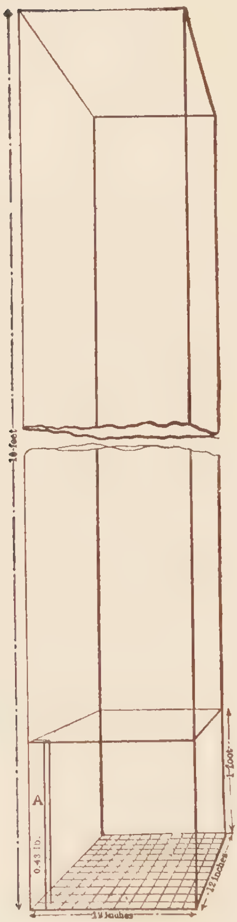

FIG. 47.

base one foot square, so that the pressure would be 620 pounds *per square foot.* The water in a tank or pond may be conceived to be divided into columns of this kind, and it will be seen that there will be a pressure on the bottom of 62 pounds per square *foot* for every foot of depth. But, in the square foot supporting this weight there are 144 square inches; and as the pressure is evenly distributed, each square inch carries:

$$62 \div 144 = 0.43 \text{ of a pound.}$$

for each foot in depth, and the pressure in the case of the column 10 feet in hight would be 620 pounds per square foot, or 4.3 pounds *per square inch.*

Just as the tank or pond could be conceived to be divided into columns of one square foot section, each square foot can be conceived to be divided into 144 columns of one square inch section, as shown in Fig. 49, and each foot in hight of such column, like the piece marked *A*, would weigh $\frac{1}{144}$ of the whole weight of the cubic foot of which it is the $\frac{1}{144}$ part, and press upon its square inch of base with a pressure of:

$$62 \div 144 = 0.43 \text{ of a pound.}$$

As this pressure in a liquid or gas is exerted in all directions, it is evident that the pressure on the horizontal piston in Fig. 50 would be 4.3 pounds *per square inch*, and if it has an area of 30 square inches there would be a force of:

$$4.3 \times 30 = 129 \text{ pounds.}$$

forcing the piston to the right; and since there is at a

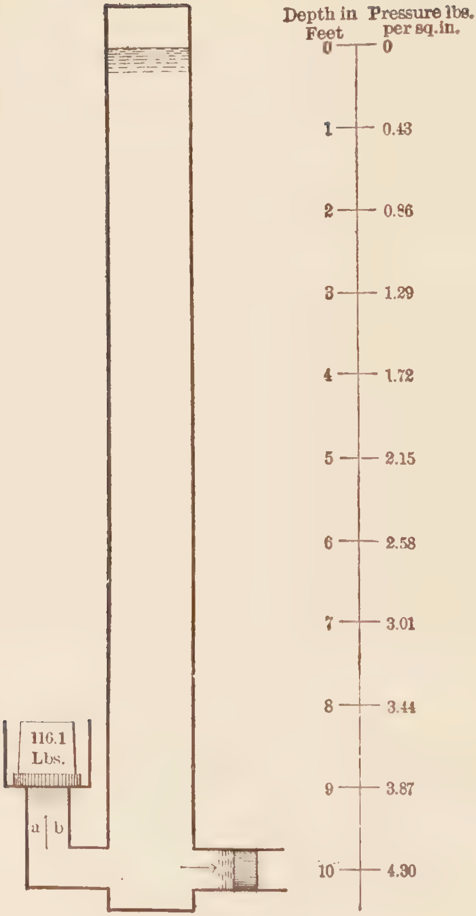
Depth in Feet
Pressure lbs. per sq.in.
0
0
1
0.43
2
0.86
3
1.29
4
1.72
5
2.15
6
2.58
7
3.01
8
3.44
9
3.87
10
4.30
116.1 Lbs.
a
b

Fig. 50.

depth of 9 feet a pressure of 3.87 pounds *per square inch* the valve at the left would have 3.87 pounds pushing upward on *each square inch* of its exposed area, *i.e.*, the area corresponding with the diameter *a b*, and if that area were 30 square inches it would take:

$$30 \times 3.87 = 116.1 \text{ pounds}$$

to hold the valve closed against that pressure.

The steam gage shows the pressure *per square inch.* If the gage points to the 100 mark it indicates that if the pressure existing in the boiler were exerted upon one square inch of area, Fig. 51, it would push with a force of one hundred pounds. If exerted upon an

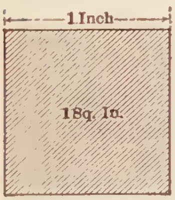

FIG. 51.

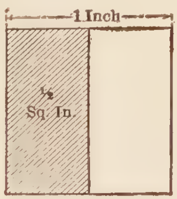

FIG. 52.

FIG. 53.

area of one-half a square inch, Fig. 52, it would push with a force of 50 pounds; upon an area ½ inch square, or ¼ of a square inch, Fig. 53, 25 pounds; upon an area of one square foot, or 144 square inches, 14,400 pounds, etc.

The force exerted by the steam to lift a safety valve depends then upon the area of the valve as well as upon the intensity of the pressure.

To Find the Area of a Circle

The area of a 1-inch circle is 0.7854 of a square inch, the difference, 0.2146, between this and the full square of the diameter being taken up by the corners, Fig. 54. If the side of the square is *doubled* the area

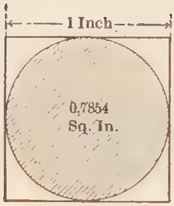

Fig. 54.

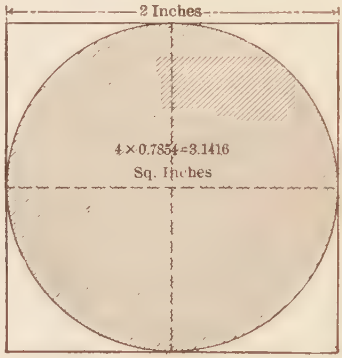

Fig. 55.

of the square will be multiplied by *four*, as is plainly shown by Fig. 55, which obviously contains *four* squares of the area of that shown in Fig. 54, although its side is but *twice* as long; and it is equally evident that the inclosed circle bears the same proportion to the total area in both cases and that the shaded area of the circle in Fig. 55 is *four* times that in Fig. 54, although its diameter is but *twice* that of the smaller circle. If we *treble* the length of the sides the area of the square will be multiplied by *nine*, always the *square* of the side, *i.e.*, the side multiplied by itself.

The area of any circle may be found by multiplying the area of a 1-inch circle (0.7854) *square inch by the square of the given diameter.*

In Fig. 55 the diameter is 2 inches and the area is:

$$2 \times 2 \times 0.7854 = 3.1416 \text{ square inches.}$$

The area of a 4-inch circle would be:

$$4 \times 4 \times 0.7854 = 12.5664 \text{ square inches.}$$

It may aid in remembering the factor 0.7854 to know that it is one-fourth of 3.1416, the number by which the diameter is multiplied to get the circumference.

The area of a triangle is obviously one-half the product of its base and hight. In Fig. 53*a* the product

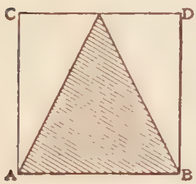

FIG. 53*a*.

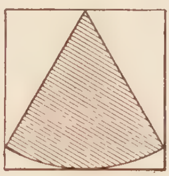

FIG. 53*b*.

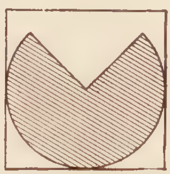

FIG. 53*c*.

of the base *A B* and the hight *A C* would be the area of the rectangle *A B C D* and the shaded area of the triangle is obviously one-half of this, for the two unshaded portions put together would make a similar triangle. This is just as true if the base is an arc of a circle as in Fig. 53*b*, and just as true if the base incloses the apex as in Fig. 53*c*. The circle is therefore a triangle with a circular base 3.1416 times the diameter

or 2×3.1416 times the radius, and with a hight equal to the radius, and its area (one-half the product of hight and base) is:

$$\text{Area} = \frac{\text{radius} \times 2 \times 3.1416 \times \text{radius}}{2} = 3.1416 \text{ radius}^2,$$

so that the area equals 3.1416 times the square of the radius, and since the radius is one-*half* the diameter, the square of the radius is the square of the diameter divided by *four:*

$$\text{Area} = 3.1416\, r^2 = 3.1416\, \frac{D^2}{4} = D^2 \times \frac{3.1416}{4}$$
$$= 0.7854\, D^2.$$

Effect of Pressure in Lifting a Valve

Suppose the 3-inch valve in Fig. 56 to be loaded with six weights of 100 pounds each and that the valve and steam weighed 30 pounds, what would the pressure per square inch have to be to lift it?

The total weight to be lifted is 630 pounds. The total upward pressure must equal this, and if 630 pounds is exerted on 7.0686 square inches (the area of a 3-inch valve, see table) the pressure on each square inch will be:

$$630 \div 7.0686 = 89.1 \text{ pounds.}$$

How much load would have to be put upon the same valve to allow it to blow off at 75 pounds per square inch?

If the pressure exerts 75 pounds on one square inch, it would exert on the 7.0686 square inches of the valve which is exposed to it:

$$75 \times 7.0686 = 530 \text{ pounds},$$

which must be the combined weight of the valve and the weights with which it is loaded.

Fig. 56 does not show a practicable valve, but is sufficient to illustrate the point that the force tending

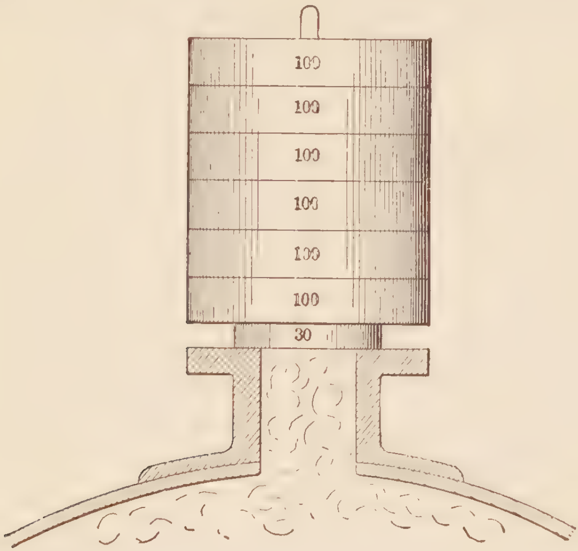

FIG. 56.

to lift the valve must equal that holding it to its seat (in this case the dead weight of the valve itself and the weights with which it is loaded), and that this upwardly acting force is the area of the valve in square inches, multiplied by the pressure per square inch. Such a dead-weight valve is ponderous and impracticable

and the usual practice is to use a lighter weight, increasing its effect by leverage, or to hold the valve to its seat with a spring.

The Principle of the Lever

Suppose a strip of board balanced over a sharp edge as in Fig. 57. If equal weights be placed upon it at

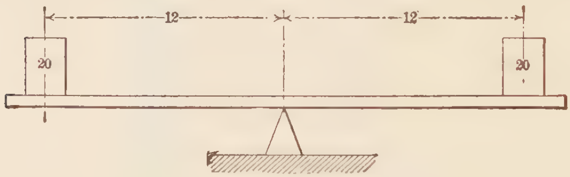

Fig. 57.

equal distances from the center it will still be in balance. If one of the weights be moved in *half* of the distance to the point at which they are balanced, as in Fig. 58, the other weight will have to be *halved* to

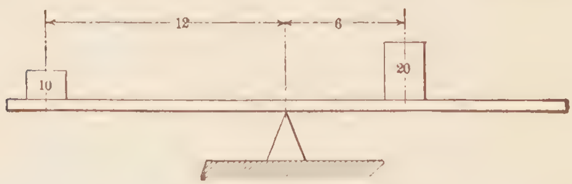

Fig. 58.

preserve the equilibrium. If one of the weights be moved to *one-third* of its distance from the balancing point, as in Fig. 59, the other weight will have to be

reduced to *one-third* of its original magnitude to preserve the balance at the original distance.

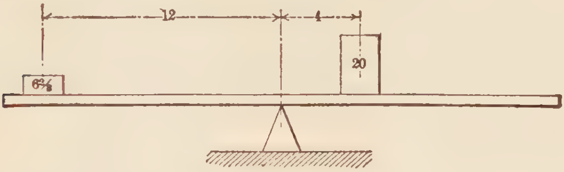

FIG. 59.

Notice that in each case the product of a weight by its distance from the point over which they balance is the same as the product of the weight which balances it and its distance from the same point. Suppose the weights in Fig. 57 to be each 20 pounds, each at 12 inches from the center. Here obviously the weights and distances being the same their products are equal:

$$20 \times 12 = 240 \quad \textit{and} \quad 20 \times 12 = 240.$$

When the right-hand weight is moved in to 6 inches from the center the other had to be reduced to 10 pounds:

$$10 \times 12 = 120 \quad \textit{and} \quad 20 \times 6 = 120.$$

When the left-hand weight was moved in to 4 inches from the center the other had to be reduced to $6\frac{2}{3}$:

$$6\tfrac{2}{3} \times 12 = 80 \quad \textit{and} \quad 20 \times 4 = 80.$$

The same principle applies in Fig. 60, where the force exerted by the man, multiplied by the distance $A\ B$, must, if he lifts the machine, equal the pressure with which the load bears on the bar at the point C,

multiplied by the distance $B\,C$ of that point from the point B around which the lever turns. In mechanics, this point, the B of Fig. 60, is called the "fulcrum" and the product of the load, weight or force by its distance from the fulcrum is called its "moment." In the case described by Fig. 57 the moment of each

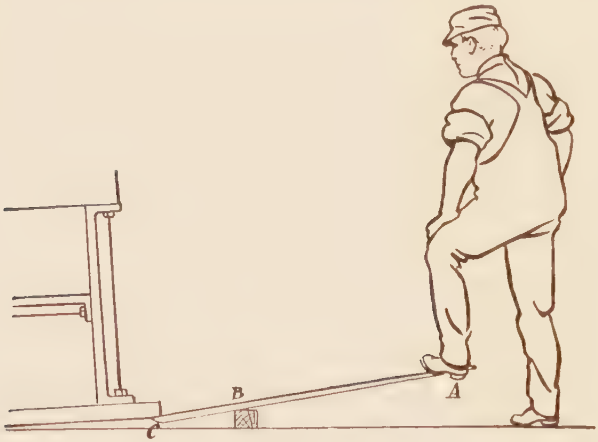

FIG. 60.

weight is 240; in that of Fig. 58, 120; in that of Fig. 59, 80; in that shown in Fig. 60 the moment of the load is the weight or force with which the load bears on the point C, multiplied by its distance from the fulcrum B, and the moment of the force is the force which the man exerts upon the bar at A, multiplied by the distance of that point from the fulcrum.

Notice that in Fig. 61 the fulcrum is at one end of

the lever instead of between the load and force as in the other examples. The principle is the same. The fulcrum is the *stationary* point about which the load and the force move. In Figs. 60 and 61 it is evident that the shorter the distance between the load and the fulcrum the less the man will have to exert himself.

FIG. 61.

The point to grasp and remember is that the *moments must be equal* in order for the force to balance or lift the load.

Equal Moments Produce Equilibrium

There are four important things about a lever:

L = the load.
F = the force applied to balance or overcome the load.
D_l = distance of the load from the fulcrum.
D_f = distance of the force from the fulcrum.

If any three of these are known the third can be easily determined, for, as has been just explained,

Force × distance of force = load × distance of load.

$$F \times D_f = L \times D_l$$

Moment of force = moment of load.

To find the force required to lift a given load:

FORMULA:
$$F = \frac{L \times D_l}{D_f}.$$

RULE. — *Multiply the load by its distance from the fulcrum, and divide by the distance at which the force is applied from the fulcrum.*

To find the distance at which a given force must be applied from the fulcrum to balance a given load:

FORMULA:
$$D_f = \frac{L \times D_l}{F}.$$

RULE. — *Multiply the load by its distance from the fulcrum and divide by the given force.*

To find the load which may be lifted with a given force:

FORMULA:
$$L = \frac{F \times D_f}{D_l}.$$

RULE. — *Multiply the given force by the distance of its point of application from the fulcrum and divide by the distance of the load from the fulcrum.*

To find the distance at which a given weight or load must be placed from the fulcrum to balance a given force:

FORMULA: $$D_l = \frac{F \times D_f}{L_l}.$$

RULE. — *Multiply the given force by the distance of its point of application from the fulcrum and divide by the load.*

THE LEVER SAFETY VALVE

Effect of the Leverage of the Ball

Suppose the weight instead of setting directly upon

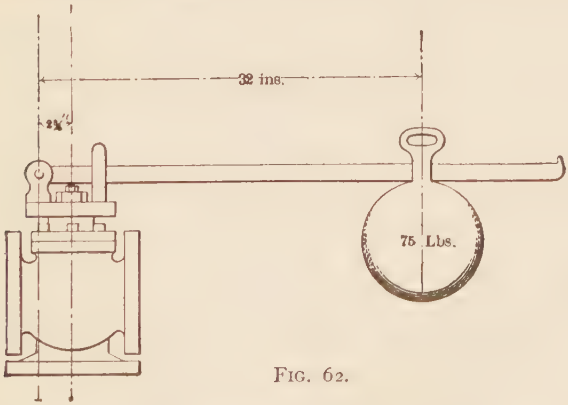

FIG. 62.

the valve, as in Fig. 56, is applied through a lever, as in Fig. 62. From what has preceded it will easily

be seen that the weight multiplied by its distance from the fulcrum will equal the force which it will exert upon the valve stem multiplied by the distance of its point of application from the fulcrum.

$$\left\{\begin{matrix}\text{Weight}\\\text{of}\\\text{ball}\end{matrix}\right\} \times \left\{\begin{matrix}\text{Distance}\\\text{of ball}\\\text{from}\\\text{fulcrum}\end{matrix}\right\} = \left\{\begin{matrix}\text{Pressure}\\\text{of ball}\\\text{on}\\\text{stem}\end{matrix}\right\} \times \left\{\begin{matrix}\text{Distance}\\\text{of stem}\\\text{from}\\\text{fulcrum}\end{matrix}\right\}$$

Let the weight equal 75 pounds,
distance of weight from fulcrum 32 inches,
distance of stem from fulcrum $2\frac{3}{4}$ inches,
what will be the force exerted by the ball to hold the valve to its seat?

$$\frac{\text{Weight of ball} \times \text{Distance of ball from fulcrum}}{\text{Distance of stem from fulcrum}} = \text{Pressure of ball on stem.}$$

Then the moment of the ball is:

$$75 \times 32 = 2400 \text{ inch-pounds,}$$

and:

$$2400 \div 2.75 = 872.727 \text{ pounds}$$

will be the pressure on the valve stem due to the ball and the moments will be equal:

$$75 \times 32 = 2400 \quad \text{and} \quad 872.727 \times 2.75 = 2400.$$

Suppose this to be a 4-inch valve, the area of which is 12.5666 square inches. The pressure per square inch upon the under side of the valve necessary to balance the effect of the ball would be:

$$872.727 \div 12.5666 = 69.4 \text{ pounds.}$$

This is the pressure at which the valve would blow off if nothing but the ball were holding it to its seat. It takes a little additional pressure to lift the valve and to overcome the weight of the lever, as will be explained later, but this is a comparatively small affair and in usual approximate calculations is not taken into account. Neglecting these we can make the following simple

Rules for lever safety valve, neglecting weight of valve, stem and lever:

Let W = weight of the ball,
D = distance of ball from fulcrum,
A = area of valve in square inches,
d = distance of stem from fulcrum,
P = pressure per square inch on valve which will balance ball.

To determine the pressure on a valve of given diameter required to balance a ball of given weight at a given distance from the fulcrum.

FORMULA:

$$P = \frac{W \times D}{A \times d}.$$

RULE. — *Multiply the weight of the ball by its distance from the fulcrum. Multiply the area of the valve in square inches by the distance of its stem from the fulcrum. Divide the first product by the second and the quotient will be the pressure per square inch required to overcome the weight of the ball.*

EXAMPLE. — The stem of a 4-inch safety valve is 2¾ inches from the fulcrum. Supposing the valve will blow when the gage shows 7 pounds without any weight upon the lever (*i.e.*, that it takes 7 pounds per square inch on the area of the valve to overcome its own weight, that of the stem and the bearing effect of the empty lever), at what pressure would it blow with a weight of 75 pounds (Fig. 62) 32 inches from the fulcrum?

BY THE FORMULA:

$$P = \frac{W \times D}{A \times d} = \frac{75 \times 32}{12.5666 \times 2.75} = 69.4 + 7 = 76.4 \text{ pounds.}$$

BY THE RULE:

Area of valve	12.5666	75	Weight of ball
Distance of stem	2.75	32	Distance of ball
	628330	150	
	879662	225	
	251332		

34.558~~150~~)2400.00)69.4 pounds.

This is the pressure required to lift the ball. Adding the 7 pounds required to blow the valve without the ball, the answer would be 76.4 pounds. Scratching out the last three figures of the first product saves handling large numbers and does not materially affect the result. If we called this 34.6 (nearer right than 34.5 because the 58 rejected is over one-half) the quotient would still be 69.36.

To find the weight required to hold a given pressure on a given valve:

FORMULA:

$$W = \frac{A \times P \times d}{D}.$$

RULE. — *Multiply the area by the pressure and by the distance of the stem from the fulcrum and divide by the distance of the ball from the fulcrum. The quotient will be the weight of ball required to balance the steam pressure on the valve.*

EXAMPLE. — What weight of ball would be required to allow the valve in the above example to blow off at 80 pounds?

The ball must provide for 73 pounds per square inch, the lever valve and stem taking care of the other seven, so that $P = 73$ pounds.

BY THE FORMULA:

$$W = \frac{A \times P \times d}{D} = \frac{12.5666 \times 73 \times 2.75}{32} = 78.8 \text{ pounds.}$$

BY THE RULE:

12.5666	Area
73	Pressure
376998	
879662	
917.3618	
2.75	Distance of stem
45868090	
64215326	
18347236	

Distance of ball, 32)2522.744950(78.8 pounds.

To find the position of the weight in order that it may exert a given pressure on the stem:

FORMULA:
$$D = \frac{A \times P \times d}{W}.$$

RULE. — *Multiply the area by the pressure and by the distance of the stem from the fulcrum and divide by the weight of the ball. The quotient will be the distance at which the ball must be from the fulcrum in order to produce a given pressure on the stem.*

EXAMPLE. — If the original 75-pound weight had been used, at what distance from the fulcrum would it have had to have been placed to have allowed the valve to blow off at 80 pounds?

BY THE FORMULA:

$$D = \frac{A \times P \times d}{W} = \frac{12.566 \times 73 \times 2.75}{75} = 33.6 \text{ inches.}$$

BY THE RULE. — The product of the factor in the numerator is 2522.74495 as before, and dividing this by 75, the weight of the ball:

75) 2522.74495 (33.6 inches.

These simple rules will serve all practical purposes, especially if it is borne in mind that P represents the pressure with which the *ball only* bears upon the stem, not including the weight of the valve, lever, etc., and an allowance be made for these other effects as has been done in the examples. A general idea of what the pressure per square inch required to lift the valve, stem and lever may be is given in column 8 of the table on page 119. It is well, however, to know how to make these allowances accurately, and they will now be considered.

Effect of the Weight of the Valve and Stem

The pressure acts directly upon the valve and stem without leverage, and must exert a force to balance their weight equal simply to that weight, just as was the case in Fig. 56.

Suppose the valve and stem of a 3-inch valve to weigh 1.5 pounds, how much pressure per square inch would be required to lift the valve from its seat?

Comparing Figs. 56 and 63, it will be seen that this case is the same as the first example given in describing the earlier cut. The total pressure on the valve must be 1.5 pounds, and if 1.5 pounds is to be exerted on 7.0686 square inches, the pressure per square inch will be:

$$1.5 \div 7.0686 = 0.212 \text{ pound.}$$

Column 3 of the table on page 119 gives roughly the weights of valve and stem used on valves of the standard diameters of three makers, and in connection with column 4, which gives the pressure per square inch required to lift the valves of the given weights, serves to indicate the relative importance of this factor of the problem.

The Effect of the Lever

The weight of the lever tends to hold the valve upon its seat. It is evident that it would take a considerable pull to lift the lever of a large safety valve with a cord attached at the point at which the pin bears, as in Fig. 64, and this pull as measured upon a scale would be

the force which the valve would have to exert to push the lever up. Every successive particle in the length of the lever is acting with a different leverage, so that it

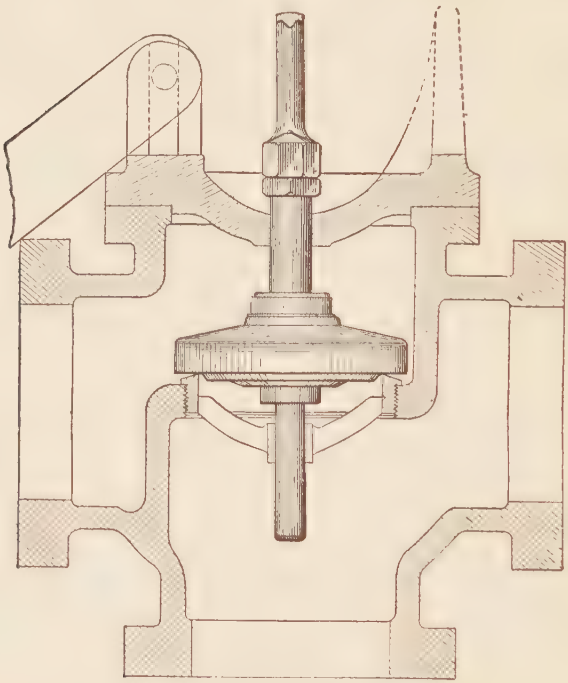

Fig. 63.

would at first appear a complicated process to calculate this force; but a body acts in this respect just as though its whole mass were concentrated at its center of gravity and this makes the problem very simple.

If the lever be taken off and balanced over an edge, as in Fig. 65, the center of gravity will be at the point

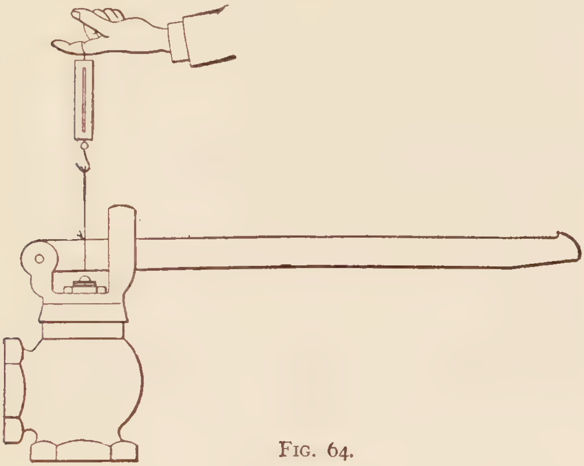

FIG. 64.

above the knife edge when the lever is balanced, and the effect of the lever would be the same as if all the mass were concentrated at that point.

FIG. 65.

Now find the distance of the center of gravity from the fulcrum, from the point around which the lever turns. This will be from the center of the hole when it turns upon a pin, as in Fig. 66, or from the point where it bears if a knife edge is used, as in Fig. 67; the distance *a c* in each case.

In measuring for moments the distances must be taken on a line passing through the fulcrum and at

Fig. 66.

right angles to the direction of the force. In the case of the lever safety valve the holding-down force is

Fig. 67.

gravity, which acts vertically. A line at right angles to the vertical is horizontal, so that distances should

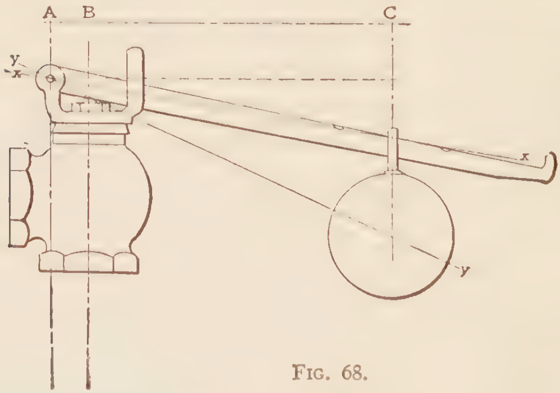

Fig. 68.

be measured in a horizontal direction as through *A B C*, Fig. 68, and not on the lines *x x* or *y y*.

In determining the distance $a\ c$, Figs. 66 and 67, do not get bothered about the piece of lever which extends back of the fulcrum. The more metal there is back of this point the nearer the center of gravity is to the fulcrum. If there were as much weight to the left of the pin in Fig. 65 as to the right, the center of gravity would be at the pin; the lever would balance over the pin as it did over the knife edge and not bear on the stem at all.

To apply this, suppose that the lever of a 3-inch valve weighed six pounds, that the distance $a\ c$, Fig. 66, between the fulcrum and the center of gravity was found to be 15 inches, and the distance $a\ b$ from the fulcrum to the point at which the pin bears $2\frac{1}{4}$ inches. The moment of the lever must be:

$$6 \times 15 = 90 \text{ inch-pounds.}$$

The moment of the lifting force must equal this, and that moment is $2\frac{1}{4}$ times the force. Then the force must be:

$$90 \div 2\frac{1}{4} = 40 \text{ pounds,}$$

$$2\frac{1}{4} \times 40 = 90 \quad \textit{and} \quad 15 \times 6 = 90.$$

Since a force of 40 pounds is to be exerted upon 7.0686 square inches, the force per square inch would be:

$$40 \div 7.068 = 5.66 \text{ pounds.}$$

The combined effect of the valve and stem and of the lever of the 3-inch valve in question would be:

$$0.212 + 5.66 = 5.87 \text{ pounds.}$$

Columns 5 and 6 of the table already referred to give

the weights of levers and the distances of their centers of gravity from the fulcrum as ordinarily found, and column 7 gives the pressure per square inch on the valve necessary to lift such levers. Column 8 gives the sum of the respective values in columns 4 and 7, *i.e.*, the pressure per square inch required to lift the valve and stem and the lever. It will be seen that the values run fairly even for all sizes of valves, and that by using seven or eight pounds as an allowance as in the above examples, results can be attained with the simple rules which will be within a pound or two of right.

Spring-loaded or Pop Safety Valves

A rule for calculating the pressure at which a spring-loaded valve will blow off is sometimes asked for. There are none reliable that do not involve the determining by experiment of the force required to compress the spring, and if you are going to do this you may as well determine by experiment at what pressure the valve will blow off. In practice nobody thinks of computing the spring-loaded valve. If they want it to blow off at 120 pounds they procure a suitable spring from the makers and turn down upon the binding nut until the valve will blow experimentally at the desired pressure. The pressure at which a spring will yield depends not only upon the shape and size of the material of which it is made, the diameter, number, and pitch of the coils, all of which are measurable and determinable, but upon the nature and condition of the material itself. You can readily appreciate that a spring of brass would compress with less pressure than one of steel, similar in

every other respect, and that there is such a wide difference in steels that there will be a great deal of difference in the action of steel springs according to the kind of metal, degree of temper, etc. The best rule known is the following:

To find at what pressure a valve will lift with a spring of given dimensions and compression:

Multiply the compression in inches by the fourth power of the thickness of the steel in sixteenths of an inch, and by 22 for round or 30 for square steel. Product I.

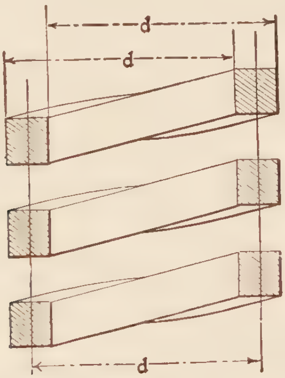

FIG. 69.

Multiply the cube of the diameter of the spring, measured from center to center of the coil (as on the line d, in Fig. 69) in inches, by the number of free coils in the spring, and by the area of the valve in square inches. Product II.

Divide Product I by Product II and the quotient will

be the pressure per square inch at which the valve will blow off.

The weight of the valve and of the spring should in strictness be added to Product I, when the construction is such that the valve supports the spring; but inasmuch as the values 22 and 30 are guessed at it will not pay to go into refinements in other directions. The result of this rule has never been compared with an actual valve. It is based on a formula adopted by a committee of Scotch engineers and shipbuilders. Correspondence with the manufacturers of pop safety valves as to the accuracy of the formula brings out the fact that they proportion and calibrate their springs only by experience and experiment. However, this rule is given for what it is worth. If you have a spring-loaded valve calculate it by this rule and see how nearly it comes to the point at which the valve will blow off.

With a dead weight or a lever-loaded valve the force required to lift it remains the same, no matter how high the valve lifts. The weights weigh no more if they are raised an inch or two, and the leverage does not change, but with the spring-loaded valve the more the valve lifts, the more the spring is compressed, and the more force is required to compress or hold it. It follows then that if an ordinary valve were loaded with a spring it would simply crack open and commence to sizzle when the pressure equaled the force at which the spring was set, and that if this were not enough to relieve the boiler the pressure would have to increase, opening the valve more and more until the steam blew off as fast as it was made.

Complete Saftey-valve Rules

It is evident that any complete rule for the safety valve must include the separate treatment of the valve and stem, the lever and the ball as factors in holding the valve to its seat.

$$\left.\begin{array}{l}\text{Moment of the}\\ \text{lifting force}\end{array}\right\} = \left\{\begin{array}{l}\text{moment of ball}\\ \qquad\qquad + \\ \text{moment of lever}\\ \qquad\qquad + \\ \text{moment of valve and stem.}\end{array}\right.$$

The lifting force consists of the pressure per square inch into the area of the valve, and its moment is the product of the force by its distance from the fulcrum. Expanded, then, the above becomes:

$$\left.\begin{array}{l}\text{Pressure}\\ \qquad \times \\ \text{Area}\\ \qquad \times \\ \text{Distance of stem}\\ \quad\text{from fulcrum}\end{array}\right\} = \left\{\begin{array}{l}\text{Weight of ball} \times \text{distance of}\\ \quad\text{its center of gravity from the}\\ \quad\text{fulcrum}\\ \qquad\qquad + \\ \text{weight of lever} \times \text{distance of}\\ \quad\text{its center of gravity from}\\ \quad\text{fulcrum}\\ \qquad\qquad + \\ \text{weight of valve and stem} \times\\ \quad\text{distance of their center of}\\ \quad\text{gravity from the fulcrum.}\end{array}\right.$$

In order to find one of these qualities we must know all the rest, and consequently since the missing quantity can be but on one side of the equal mark we can figure the combined value of the quantities on one side of the

equation (that is, in one set of brackets). Then we can work out the operation indicated on the other side as far as we can go. If the missing quantity is on the left-hand side of the equation it can be found by dividing the value of the other side of the equation by the product of the two known factors on the left-hand side.

To find the pressure at which a certain valve will blow off:

Multiply the weight of the ball, of the valve and stem and of the lever, each by the distance of its center of gravity from the fulcrum and add the products. Multiply the area of the valve by the distance of its center from the fulcrum and divide the sum above found by the product. The quotient will be the pressure required.

Or more briefly:

Divide the sum of the moments of the valve, lever and ball by the product of the area of the valve and distance from the fulcrum.

Example.—At what pressure will a 3-inch valve blow off with stem $2\frac{1}{4}$ inches from the fulcrum, valve and stem weighing $1\frac{1}{2}$ pounds, lever weighing 6 pounds, having its center of gravity 15 inches from the fulcrum and weighted with a 48-pound ball 24 inches from the fulcrum?

$$\left.\begin{array}{l}\text{Pressure}\\ \qquad\times\\ \text{Area} = 7.0686\\ \qquad\times\\ \text{Distance } 2\tfrac{1}{4}\\ \text{Product} = 15.90435\end{array}\right\} = \left\{\begin{array}{rcr} 48\times 24 &=& 1152\\ 6\times 15 &=& 90\\ 1.5\times 2\tfrac{1}{4} &=& 3.375\\ \text{Sum of moments} &=& 1245.375\end{array}\right.$$

$$1245.375 \div 15.90435 = 78.3 \text{ pounds.}$$

This is all that we shall be likely to wish to find on this side of the equation, for the distance of stem is fixed and the area determined by other considerations.

The other two things that interest us are the weight of the ball and its distance from the fulcrum.

To find weight of ball or its distance from fulcrum:

Multiply the pressure by the area and by the distance of the stem from the fulcrum. The product is the moment of the force.

Multiply the weight of the valve by the distance of the stem from the fulcrum; multiply the weight of the lever by the distance of its center of gravity from the fulcrum, and add the products.

Subtract the sum of the products just found from the moment of the force, and the difference is the moment of the ball.

Divide the moment of the ball by the weight of the ball and the quotient is its distance from the fulcrum.

Divide the moment of the ball by the distance from the fulcrum and the quotient is the weight of ball required.

EXAMPLE — What weight of ball at the same distance would be required to allow the valve given in the previous example to blow at 75 pounds, and at what distance would the 48-pound ball there given have to be placed from the fulcrum to produce the same result?

75 × 7.0686 × Moment $2\frac{1}{4}$ of force 1192.826 93.375 1099.451 moment of ball	Weight × distance of ball + $6 \times 15 = 90.000$ + $1.5 \times 2.25 = 3.375$ 93.375 sum of moments, valve and lever.

$$\frac{1099.451}{48} = \text{inches, distance of ball.}$$

$$\frac{1099.451}{24} = \text{pounds, weight of ball.}$$

But the ideal valve should stay on its seat until the pressure reaches the desired limit, then open wide and discharge the excess. This result is accomplished by the construction shown in Fig. 70. With the first opening of the valve the steam passes into the little "huddling chamber" made by the cavity near the overhanging edge of the valve and a similar cavity surrounding the seat. The pressure which accumulates here, acting on the additional area of the valve, raises it sharply with the "pop" which gives the valve its name, and it is sustained by the impact and reaction of the issuing steam until the pressure has subsided sufficiently to allow the spring to overcome these actions.

The outside edge of the lower trough in the valve shown is composed of an adjustable ring which may be

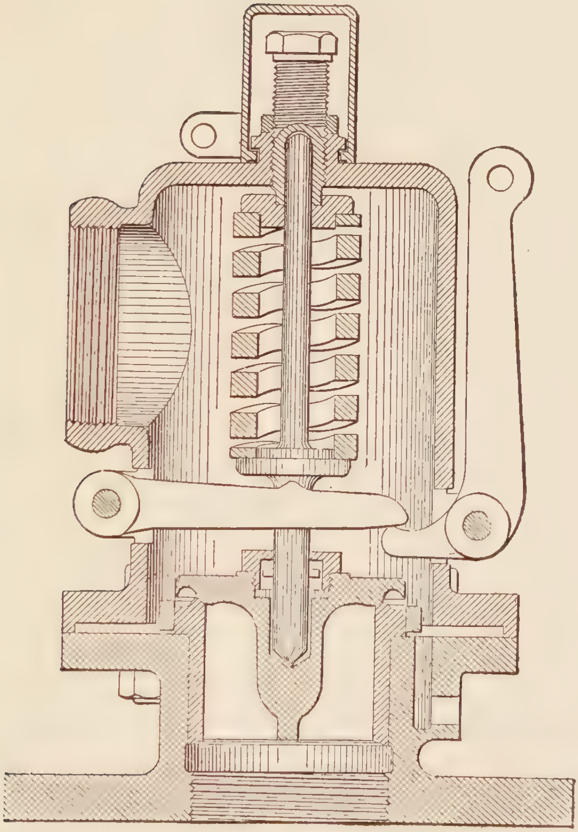

FIG. 70.

screwed up or down so as to diminish or increase the distance between the overhanging lip of the valve and its own inner edge, controlling the outlet from the chamber; and diminution of pressure or the "blow back" required to allow the valve to seat so that the valve opens wide at a given pressure and seats promptly without sizzling or chattering when the pressure has been reduced a certain amount depending upon the adjustment of the ring. The various makers have adopted different devices for adjusting the ring or other device for controlling the outflow from the huddling chamber.

The Capacity of Safety Valves

Let us next consider the capacity of valves; how large a valve is required for a given boiler. Most of the rules deal with grate surface and the area of the valve; the rule adopted by the U. S. Board of Supervising Inspectors being one square inch of valve area for each two feet of grate area. That the valve should be proportioned to the grate surface seems proper because it is the grate surface, and not the heating surface, which determines and limits the capacity of a boiler. To a given grate surface, however, we should apportion a sufficient amount of area of opening, and this area of opening is not proportional to the area of the valve but to the diameter and lift. A valve 1 inch in diameter has an area of 0.7854 of a square inch, but that does not mean that there will be an opening of 0.7854 of a square inch for the steam to escape. If the valve is flat, as in Fig. 71, the area opened for the discharge of steam

will be the circumference of the valve multiplied by the lift. The circumference is

$$\text{Diameter} \times 3.1416 \qquad (1)$$

and the area of the complete circle is

$$\text{Diameter} \times 3.1416 \times \frac{\text{Diameter}}{4} \qquad (2)$$

and the area for the escape of steam is

$$\text{Diameter} \times 3.1416 \times \text{Lift}. \qquad (3)$$

When the lift is one-quarter the diameter, or

$$\frac{\text{Diameter}}{4},$$

the area for the escape of steam is the same as the area of the circle; formula 3 is the same as formula 2.

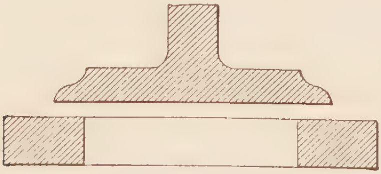

FIG. 71.

When a flat valve has lifted a quarter of its diameter it has reached the limit of its capacity to discharge steam. It doesn't do any good to lift higher, for the area around the edge of the valve is already as large as the area of the valve itself and the capacity of the valve is proportional to the area or the square of the diameter.

In practice, however, the lift of valves is much less than one-quarter of their diameter, and for a given lift the area for the escape of steam is proportional to the circumference or the diameter rather than to the area. Most of the rules, however, as above stated, allow a given amount of valve area to a square foot of grate surface, and make the allowance liberal enough to include all conditions. For instance, the rule of the U. S. Board of Supervising Inspectors calls for one-half a square inch of valve area for each square foot of grate surface. A 4-inch valve has about 12 square inches of area and would thus take care of 24 square feet of grate. It would not be possible to burn over 25 pounds of coal per square foot of grate per hour with natural draft, nor to evaporate over 12 pounds of water with a pound of coal, so that the boiler could not possibly make more than

$$25 \times 12 \times 24 = 7200 \text{ pounds of steam per hour,}$$

or

$$7200 \div (60 \times 60) = 2 \text{ pounds of steam per second.}$$

Now the weight of the steam which will escape through a given aperture per second is given by the following formula:

$$\text{Wt.} = \frac{\text{Pressure} \times \text{Area}}{70}$$

that is, the weight in pounds which will escape in a second is equal to the absolute pressure in pounds per square inch multiplied by the area in square inches and divided by 70.

On the other hand, the area required to discharge a given weight is

$$\text{Area} = \frac{\text{Weight} \times 70}{\text{Pressure}}$$

that is, the weight in pounds to be discharged per second multiplied by 70, and divided by the absolute pressure equals the required area. Now we have found that with a rate of combustion practically impossible, with natural draft, and a practically unattainable evaporation per pound of coal, the most steam that the boiler with 24 square feet of grate suface could furnish is 2 pounds per second. The area required to discharge this at 70 pounds pressure, absolute, is

$$\frac{2 \times 70}{70} = 2 \text{ square inches.}$$

The 4-inch valve which this boiler would require would have a circumference of practically 12 inches, and would need to lift only one-sixth of an inch to furnish the two square inches of opening necessary to discharge the steam, for

$$12 \times \tfrac{1}{6} = 2.$$

One-sixth of an inch is only one twenty-fourth of the diameter of the valve. You see that this simple rule gives an ample margin, requiring but a small lift to discharge more steam than the boiler can possibly make. It is altogether useless and nonsensical to figure the areas of opening to four places of decimals involving with beveled seats complicated operations with sines and cosines, in a calculation which involves no accuracy

but which requires simply a result which shall be amply large to cover any emergency likely to be encountered in practice. It is like trying to measure the distance to the next town in feet and inches, in order to answer a man who would be abundantly satisfied to know that it was about three-quarters of a mile. You may be sure that a valve which has a square inch of area for each two square feet of grate surface will liberate all the steam that can be made by the coal that you can burn on that grate surface, so long as the valve is free and in good condition. It is quite probable that a smaller valve would do, but in a matter of this kind we want to provide not the smallest that will possibly do but enough capacity to be absolutely safe. For all purposes of ordinary practice, therefore, divide the grate surface by 2, which will give you the valve area required and you can find the corresponding diameter by multiplying the square root of the area by 1.128. Don't carry your decimals out too far because you will have to take the nearest commercial size after all.

Here is a rule which will give you the diameter of the valve in inches at once:

Multiply the square root of the grate surface by 0.8.

This would be particularly handy when the grate is square, or nearly so, for then the length would be the square root of the area.

You can see how the rule is made, or rather, makes itself.

By the supervising inspector's rule the valve area required equals the grate surface divided by 2.

$$\text{Area} = \frac{\text{grate surface}}{2}.$$

The diameter is the square root of the quotient of the area divided by 0.7854.

$$\text{Diameter} = \sqrt{\frac{\text{area}}{0.7854}}.$$

And since in this case the area equals one-half the grate surface the diameter will be the square root of one-half the grate surface divided by 0.7854.

$$\text{Diameter} = \sqrt{\frac{\text{grate surface}}{2 \times 0.7854}};$$

or,

$$\text{Diameter} = \sqrt{\frac{\text{grate surface}}{1.5708}}.$$

We can get rid of the square root in the denominator by finding it once for all. It is 1.25 very nearly. So our formula becomes

$$\text{Diameter} = \frac{\sqrt{\text{grate surface}}}{1.25}.$$

Dividing by 1.25 is just the same as multiplying by $\frac{1}{1.25}$, and as $\frac{1}{1.25} = 0.8$, the multiplication is easier, so we have

$$\text{Diameter} = \sqrt{\text{grate surface}} \times 0.8.$$

The grate surface will never be so large that the square root cannot be easily determined with sufficient accuracy mentally. If it is between 25 and 36 the root is between 5 and 6. The square of 7 is 49, of 8, 64, etc., so that by trial the root can be determined approxi-

mately. Here is an easy trick to get the square of a number with two figures ending in 5:

Multiply 1 *plus the left-hand figure by the left-hand figure, and annex* 25 *to the product.*

What is the square of 35?

The left-hand figure is 3. Three plus 1 is 4, and $4 \times 3 = 12$. Annex 25 and get 1225.

This rule works just the same when the 5 is a decimal, only in that case the annexed 25 is a decimal too, and will enable you to determine instantly by inspection the nearest number advancing by halves to the square root. As the sizes of safety valves advance by half inches, the nearest root determined in this way will be sufficiently accurate, as we have to take the nearest commercial size anyhow.

What is the square of 6.5?

Six plus $1 = 7$; $7 \times 6 = 42$; add 25, which in this case will be a decimal fraction, there being two places to point off, and get 42.25.

In this way you can square 1.5, 2.5, 3.5, etc., and this is as near as it is ever necessary to get a root in the above formula. Suppose, for instance, you had 58 square feet of grate surface. What is the square root? Seven times $7 = 49$, and $8 \times 8 = 64$. It must be between 7 and 8; $7.5 \times 7.5 = 56.25$.

That is near enough to 58. The square root of 58 is really 7.615. Multiplying this by 0.8 we get $7.615 \times 0.8 = 6.092$, which is practically a 6-inch valve. We should have got at the same result if we had taken the square root as 7.5, for $7.5 \times 0.8 = 6$.

When the grate surface is over 30 or 40 feet it is

better to get the required capacity by putting on two valves than by using one large one. In fact it is a pretty good plan to have two safety valves anyway. There is a great deal of responsibility on that little appliance, and many of the most destructive of boiler explosions would have been avoided by an operative safety valve of sufficient capacity. So many little things can occur to make it hold against a destructive pressure, even when the attendant follows the usual directions to raise it from its seat daily, that prudence dictates the use of an auxiliary valve. It would be a remarkable coincidence if both stuck at the same time without criminal negligence.

The amount of opening of an ordinary lever safety valve is determined by the amount of surplus steam to be delivered. If the boiler is making more steam than is to be taken out of it the pressure will increase, and when it reaches an amount sufficient to overcome the weight of the ball, etc., the valve will be raised a little from its seat and the steam will escape. If the opening thus afforded is sufficient with the other drafts on the boiler (such as the supply to the engine, etc.) to allow all the steam the boiler is making to escape, the valve will not open any wider, but if not the pressure will continue to increase and force the valve open until the steam can escape as fast as it is made. As the surplus production of steam decreases, as by closing the dampers or a greater demand by the engine, the valve gradually settles down to its seat again.

On account of its greater lift and effective discharging area the pop valve is allowed by the Board of Super-

vising Inspectors three square feet of grate surface per inch of area instead of two, as with the ordinary lever valve.

We have seen that the escape of steam through an opening of given size is proportional to the absolute pressure. Twice as much steam will go out of an inch hole in a minute with 190 pounds behind it as with 95 pounds. It is presumed, for this reason, that the inspectors only require a square inch of valve area for every 6 feet of grate surface on boilers carrying a steam pressure exceeding 175 pounds gage.

It has been said that although the area effective for the escape of steam is not proportional to the area due to the diameter of the valve, and although the latter area is that used in the formula for capacity, the allowance is so liberal that it is practically useless to figure the former. It may be interesting, however, to know how to figure it, and a treatise on the safety valve would hardly be complete without directions for so doing.

With a flat valve we have already seen that the area for the escape of steam is the lift of the valve multiplied by its circumference. With a bevel-seated valve in which the valve does not lift out of the seat the area *A A*, Fig. 72, is that of a frustum of a cone, Fig. 73. Now to find this area the rule is to add the circumference of the greater circle to the circumference of the lesser *C D*; divide by 2, and multiply by the slant hight *C A*. In other words, to multiply the average length of the strip which would be made by flattening this surface out by the width of that strip. To work this

rule out would take us too far into trigonometry, but the rule follows:

(1) *Multiply the diameter of the valve by the lift, by the* ***sine*** *of the angle of inclination and by* 3.1416.

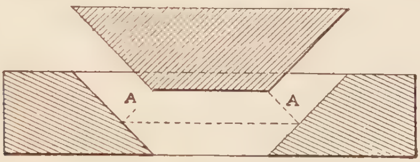

FIG. 72.

(2) *Multiply the square of the lift by the square of the sine of the angle of inclination, by the cosine of this angle and by* 3.1416.

(3) *Add these two products.*

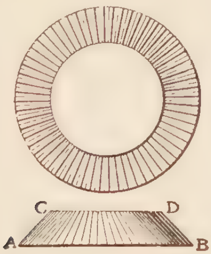

FIG. 73.

The U. S. rules require a bevel of 45 degrees, and most valves are made with seats of that degree of inclination. For such a valve the rule becomes:

(1) *Multiply the diameter of the valve by the lift and by* 2.22.

(2) *Multiply the square of the lift by* 1.11.

(3) *Add these two products.*

When a valve with a beveled seat lifts clear of the seat as a valve with a slight bevel may, the area of the opening is computed by the above rule for a lift which would raise it to the upper level of the seat, and to this is added the circumference of the valve multiplied by the lift above the seat level.

1	2	3			4		
Diam'r of Valve	Area of Valve	Weight of Valve and Stem			Pressure Required to Lift Valve and Stem		
In.	Sq. In.	Pounds			Pounds per Sq. In.		
$\frac{3}{8}$	0.1104	0.125			0.131		
$\frac{1}{2}$	0.1963	0.156		0.14	0.7947		0.713
$\frac{3}{4}$	0.4418	0.187		0.23	0.423		0.521
1	0.7854	0.187		0.34	0.238		0.432
$1\frac{1}{4}$	1.2272	0.312		0.60	0.254		0.488
$1\frac{1}{2}$	1.7671	0.437		0.75	0.247		0.424
2	3.1416	0.542	1.5625	0.97	0.172	0.497	0.308
$2\frac{1}{2}$	4.9087	0.8395	2.75	1.69	0.171	0.560	0.344
3	7.0686	1.339	3.50	2.33	0.189	0.495	0.329
$3\frac{1}{2}$	9.6211	1.8	4.75	2.60	0.187	0.494	0.270
4	12.5666	2.371	5.75	4.12	0.189	0.458	0.327
$4\frac{1}{2}$	15.9043	3.0	6.75	5.18	0.189	0.424	0.326
5	19.6350	4.125	9.75	6.45	0.210	0.497	0.324
6	28.2744	5.87	11.875	8.62	0.208	0.420	0.305

5			6			7			8		
Weight of Lever			Distance of Center of Gravity from Fulcrum			Pressure Required to Raise Lever			Pressure Required to Raise Valve, Stem and Lever		
Pounds			Inches			Pounds per Sq. In.			Pounds per Sq. In.		
0.125			3.25			5.89			6.003		
0.140		0.20	3.0		6.25	2.85		8.49	3.6447		9.203
0.343		0.38	4.812		9.0	4.98		10.32	5.403		10.841
1.0		0.48	7.75		9.0	8.32		5.50	8.558		5.932
1.125		0.65	7.312		8.8125	5.65		3.73	5.904		4.218
0.875		0.87	6.875		10.125	2.87		3.63	3.117		4.054
2.5	3.0	1.12	13.312	14.56	11.50	7.37	7.42	2.43	7.542	7.92	2.738
3.5	3.562	4.0	15.25	16.37	15.0	5.79	5.59	7.24	5.961	6.15	7.584
4.75	5.812	6.0	18.625	17.37	19.25	6.67	6.35	7.07	6.859	6.84	7.399
5.75	8.25	10.50	21.75	19.0	19.25	5.78	6.52	9.08	5.967	7.01	9.350
6.0	12.875	13.0	23.0	22.0	23.25	4.78	8.20	8.75	4.969	8.66	9.077
7.0	13.125	18.0	22.125	23.0	26.75	3.89	6.33	11.09	4.079	6.75	11.416
10.75	18.25	20.0	25.25	25.5	31.25	5.53	7.22	10.62	5.74	7.72	10.944
15.0	21.25	32.0	27.5	29.62	37.25	5.56	6.36	12.05	5.768	6.78	12.355

9			10					
Length of Lever			Distance of Stem from Fulcrum			Weight of Ball Ordinarily Furnished		
Inches			Inches			Pounds		
6.31			0.62			1.56		
5.62		12.5	0.75		0.75	3.2		2.0
9.50		18.0	0.75		0.75	5.5		2.6
14.87		18.0	1.19		1.0	8.12		4.8
14.4		17.625	1.19		1.25	9.62		11.0
13.4		20.25	1.19		1.37	15.37		14.0
26.1	29.50	23.0	1.44	1.87	1.687	19.0	30	24.0
30.0	33.0	30.0	1.87	2.12	1.687	29.0	45	34.5
37.12	35.0	38.5	1.87	2.25	2.312	38.0	63	50.5
43.1	38.375	38.5	2.25	2.50	2.312	48.5	88	67.0
45.5	44.5	46.5	2.37	2.75	2.75	70.0	110	86.5
43.75	46.125	53.5	2.5	3.0	2.75	83.0	140	86.5
49.87	51.50	62.5	2.5	3.25	3.0	98.0	168	103.0
54.5	59.50	74.5	2.625	3.50	3.5	139.0	220	139.0

X

HORSE-POWER OF BOILERS[1]

In a recent catalog of a well-known maker of engineering specialties the following approximate rules for calculating the horse-power of various kinds of boilers were noticed and copied. The rules are intended for use in determining the proper sizes of injectors and other apparatus when the exact dimensions or heating surface of the boiler is unknown or hard to obtain:

KIND		H. P.
Horizontal Tubular	=	$\text{Dia.}^2 \times \text{Length} \div 5$
Vertical Tubular . .	=	$\text{Dia.}^2 \times \text{Hight} \div 4$
Flue Boilers	=	$\text{Dia.} \times \text{Length} \div 3$
Locomotive Type	=	$\text{Dia. of Waist}^2 \times$ Length over all $\div 6$.

All dimensions to be in feet.

In the first and third cases the length is the length of the tubes or that of a "flush-head" boiler and does not include the extended smoke-box. In the second case, the hight is that of a plain vertical boiler in which the upper part of the tubes is above the water line; it is not the hight of a boiler with submerged tubes.

[1] Contributed to Power by C. G. Robbins.

The extreme simplicity of the rules aroused curiosity as to their accuracy, and comparisons were made between manufacturers' ratings and ratings calculated by the formulas above. The results are given in the accompanying table. They agree very closely, except in a few of the larger sizes of tubular boilers, where the calculated rating falls below that of the manufacturer. And in these sizes it will be noticed that the heating surface per horse-power is less than in the smaller sizes where the two ratings practically agree.

It is quite possible that the ratings of other manufacturers would show a better or worse agreement. In any event, the rules prove to be valuable for just what is intended and will save considerable trouble in measuring up and calculating the power of existing boilers when ordering injectors, feed pumps, and the like.

HORIZONTAL TUBULAR

Diameter (inches) × Length (feet)	36×8	36×10	42×10	42×12	48×12	54×13	54×15	60×14
Heating Surface	232	290	370	429	568	684	800	944
Manufacturers' Rating	15	20	25	30	40	50	60	70
Rating by Rule	14.4	18	24.5	29.4	38.4	52.7	60.8	70

HORIZONTAL TUBULAR (CONCLUDED)

Diameter (inches) × Length (feet)	60×15	60×16	66×15	66×16	72×16	72×18	78×18	84×18
Heating Surface	1010	1080	1249	1350	1674	1829	2115	2511
Manufacturers' Rating	75	80	90	100	125	150	175	200
Rating by Rule	75	80	90.8	96.8	115.2	130	152.1	176.4

VERTICAL TUBULAR

Diameter (inches) × Hight (feet)	24×5	30×5	30×6	36×7	40×7½	44×8	48×8	48×9	54×9
Heating Surface	58	84	100	180	238	292	358	480	578
Manufacturers' Rating	5	7	8	15	20	25	30	40	50
Rating by Rule	5	7.8	9.4	15.6	20.6	26.8	32	36	45.6

TWO-FLUE BOILERS

Diameter (inches) × Length (feet)	36×18	36×22	42×20	42×24	44×26	48×26	40×30	50×32	54×34	60×33
Heating Surface	206	253	285	338	393	415	478	542	610	663
Manufacturers' Rating	15	20	25	30	35	40	45	50	60	70
Rating by Rule	18	22	23.3	28.7	31.8	34.7	40	44.5	51	55

LOCOMOTIVE TYPE

Waist Diam. (in.) × Length (ft.)	40×14	42×14⅓	44×15	44×16⅓	48×17½	54×18½	56×19½	58×19¾	60×21	62×22¼
Heating Surface	268	309	350	400	503	596	710	809	898	1050
Manufacturers' Rating	25	30	35	40	50	60	70	80	90	100
Rating by Rule	25.9	29.3	33.6	36.6	46.7	62.5	72	77	87.5	90

XI

BOILER APPLIANCES AND THEIR INSTALLATION

In this paper it is my aim to briefly point out a few of the deficiencies which not only exist in so many of the less modern plants located in isolated places, but which are too often found in the large and perhaps otherwise well-equipped plants.

First in importance is the safety valve, which in some instances can be called such in name only, for in their neglected or overloaded condition they would not in any sense answer the purpose for which they are intended. We still find a few engineers who persistently stick to the old-style lever and weight safety valve — for what reason, we cannot say, unless it is because they can be overloaded more easily than the more modern spring-loaded pop valve. Certainly everything else is in favor of the pop valve, especially in the hands of incompetent persons, for the most successful design of any steam appliance is that one which is absolutely fool-proof. From the fact that they can be locked and made fool-proof, that they are much more reliable in their action and so much less wasteful of steam, it is believed they will soon be used universally, and that the lever valve will be a

thing of the past. But with all their advantages they will not relieve the boiler of over-pressure unless properly installed and kept in operative condition.

Boilers have been seen equipped with these valves ample in size to take care of all the steam the boiler could generate, and then the discharge opening reduced to one-third the area of the valve and piped up through the roof. Again, as many as four 72×18 boilers have been seen all equipped with 4-inch pop valves, and all piped to blow into one continuous 4-inch header, which extended through the wall. There is no serious objection to piping the waste steam from a safety valve out of the building, when properly done, but the better plan would be to have a suitable ventilator in the roof, and let them discharge in the building. There are, however, many plants where this cannot be done. If the waste pipe is run out of the building, it should never be smaller than the valve itself, and if it is necessary to carry it any great distance, 20 feet or more, the pipe should be increased one size and connected up with as few turns as possible. It is also a very dangerous plan to run a waste pipe direct from the safety valve horizontally some distance, and then run a vertical pipe up through the roof, unless the pipe is properly supported to not only sustain its own weight, but to carry the downward thrust due to the reaction of the steam, which would in turn throw a severe strain on the casing of the valve and the flange bolts. The amount of pressure so exerted is of course a matter of conjecture, for the full boiler pressure could hardly be expected to be

realized on the waste pipe. However, serious accidents are known to have happened from just such construction, therefore they are not mere possibilities. It is also quite necessary that the waste pipe be supplied with the proper opening for free and continuous draining, and not depend altogether on the drip opening on the valve itself.

Next in importance to the safety valve is the water column and its connections. There are probably more accidents to boilers traceable to defective water columns than to any other one cause. On a recent visit to a new plant where three 150 horse-power boilers were being installed, the water columns were found piped up with $\frac{3}{4}$-inch pipe, with several turns in the lower connection and with no blow-off pipe. With some feed waters this would probably answer the purpose, but water used for boiler purposes is often found which would close up the lower connection in a very few days' run. In this case, as in many others, the boiler makers were at fault, as they were furnishing the attachments. Water columns should never be connected up with pipes smaller than $1\frac{1}{4}$ inches, and in general practice $1\frac{1}{2}$-inch pipe is better, but in every instance the lower connection should be provided with a $\frac{3}{4}$-inch blow-off, and for convenience should be piped to discharge into the ash-pit. This blow-off may be provided with any good valve or cock, but if the latter is used, a closed end wrench should be provided, as an adjustable wrench is too apt to be carried away and the blowing out neglected. The removable disk Y-valves now on the market have

been found very serviceable and reliable for boiler blow-offs, and no doubt they would be equally as good for water columns. There is quite a difference of opinion among engineers as to the advisability of placing stop valves in the column connection, but there is no real good reason advanced why they should be so equipped. Many plants with valves in both the lower and upper connections are found, but it is not a misstatement to say that one-half the lower valves can be found in an inoperative condition, owing to the accumulation of scale on the seat and valve. The less the number of attachments which may prove a source of danger the better. In connecting up water columns it is a good plan to use crosses in the lower connections, plugging the unused openings with gun-metal or brass plugs. These will be found very convenient for removing deposit which may accumulate and cannot readily be blown out. Water columns are often too small to give the best results. The chamber should be at least 3 inches, and preferably 4, in diameter, internally.

Ignorance is also often displayed in placing water-columns. A column placed too high is fully as dangerous in the hands of some men as one placed too low. In plants with the columns so placed it has been observed that when the water was just visible in the bottom of the glass there would be 6 inches of water above the top of upper row of tubes. The fireman, knowing this fact, will carry the water low in the glass, and if by chance the water should disappear from view altogether, he will tell you that it just went out

of sight, and will then proceed to speed up the boiler feed pump. A better and safer plan is to set the column with the bottom of the glass just level with the top of the upper row of tubes, then pull or cover the fire before the water leaves the glass, in case of the failure of the water supply. Gage-glass valves and try-cocks are also too often neglected. While it is believed there is no better way of ascertaining the hight of water in boiler than by blowing out the water column and then noting the rapidity with which water returns to the glass, do not neglect the try-cocks, for the water glasses will break and at times when it is not convenient to replace them; then the try-cocks will come in handy, and should be found in working order.

The steam gage is next in importance, but, as a rule, receives little attention. The dial on the factory clock is kept clean, so there will be no mistake in reading the time when the whistle should be blown; but with the gage it is different; it has no such important duties to perform. A steam gage is not the delicate instrument that some would believe. However, their accuracy is easily destroyed, if not properly connected up. They should not be attached to the breeching or boiler front, unless protected from the heat, and they should be provided with a water trap to protect the Bourdon spring from the heat of the live steam; otherwise they will not give correct readings, and may be ruined altogether. The best form of trap is one made up of nipples and fittings, with a small drain cock placed in the lowest point of trap for the purpose of blowing out and of draining, to prevent freez-

ing in winter. A trap of this kind will be found much easier to clean, in case of stopping up, than the bent pipe.

Notes

The use of cast-iron flanged nozzles connecting boilers to steam pipes is being superseded by dropped forge steel flanges. The former are objectionable, for the rivet holes must be accurately drilled and the curve must be a neat fit to the boiler plate or a calking gasket be provided to make the joint tight. Then such flanges will fail by fracture in riveting after some time and money has been spent on the job. If the cast nozzle is flanged to receive the steam pipe, the holes for bolts must be drilled out and bolts go with the nozzle. Many buyers object to superfluous flanges as that much more for the engineers to care for and pack. Then if the plant be a one or two boiler one the best plan is to have no flanges between the boiler and the main steam valve, for if a flange blows out the packing one can readily shut the main steam valve and repack it, while on the other hand one will wait until the boiler cools off. With regard to strength of material, while cast iron can be made, and undoubtedly is, to run as high as 25,000 pounds tensile strength, the fact is it may run less than that, and in calculating averages must be taken, which doubtless will not exceed 15,000 pounds per square inch. In addition allowance must be made for shrinkage strains in such castings; usually an unknown amount, but allowance should be given. If the flange be threaded the strength of such threads will of course not equal

threads in forged steel. The expansion of the boiler shell sets up strains on these flanges, and while cast iron resists bending or flexure well, that is of no special value, as the strains are continuous and unavoidable. On the contrary dropped forged steel flanges furnished in all pipe sizes and to fit practically any required circle are now made and kept in stock by boiler supply houses, ready for attaching and tapped to size.

The tensile strength equals flange steel. Granting cast iron 15,000 tensile strength and forgings 60,000, note the wide difference in strengthening by reinforcement of the hole cut in the boiler shell. The steel flange may have rivet holes punched to within $\frac{1}{8}$-inch of size and reamed out without damage. It will "give" in fitting and riveting to the shell and no gasket is needed. A Fuller calking tool will quickly close any leak at the seam, but such rarely occurs, as it is forged on a smooth die. The threads are $1\frac{1}{2}$ inches deep on a 6-inch size, and are very strong. From all of the above, proved by experience, the steel flange is vastly better than the cast-iron one. Nevertheless many old men will not accept anything other than the latter, doubtless due to their opinions having formed and "froze in" years ago.

The foregoing applies to a large extent to cast-iron man-head frames, but as such are of larger size it is clear that the strains due to shrinkage and expansion under working pressure must be kept in mind as the casting is narrower at the cross-section of the minor ellipse than at major. Presuming the casting projects upward and inward to receive the plate, the usual

type, it is likely shrinkage strains occur at the corners of the angles. Considering the large amount of the shell cut away for an 11×15 inch man-head, and allowing 15,000 tensile strength for castings, it is often the case that the weakest link in the chain composing the strength of a boiler is at this point. In my opinion engineers, designers and boiler makers should abandon cast-iron man-head frames, if for no other reason than to strengthen the boiler. Such frames should be, as high-class shops now use, weldless flange steel, forged into shape and double-riveted to the shell plate. No one thus far has seen a cast man-head frame double-riveted to the shell. Please note that point.

With an elliptical steel frame to save packing and also profanity, a ½-inch ring should be shrunk on to the inner flange and planed off to give a seat one inch wide. But if packing costs money, then in the man-head plate have a groove provided so the packing cannot be forced out, and a piece of asbestos ¾ rope with plumbago and oil when the joint is opened will last two years. In one case with two boilers washed out every two weeks it cost \$3.00 each opening for gaskets. The old plates were on my advice replaced with new ones grooved and the cost for gaskets reduced from \$78 per annum to \$2, a thrifty saving by the way.

Returning to the strength of the steel frame, it is so superior to the cast one in every manner that no comparison can be made. Nor indeed is it necessary to buy one particular type, as while these frames are patented, several being of about equal merit, prices

are not kept at a high point. In view of the failures in riveting castings and including drilling it is doubtless true the steel frames like the steel nozzles are cheaper to the builder, while in every way each should be more desirable to the buyer, the engineer and the insurance companies.

To a large extent the above applies with equal truth to pressed steel lugs or brackets supporting the boiler, but in addition in shipping a boiler, while a steel lug may by transit be bent, it can easily be straightened and without injury — a valuable quality. When a boiler is set, the lugs being out of sight are out of mind. As the walls transmit heat it is clear the lugs become quite warm, for it is usual to protect them by the thickness of only one brick, and as the lug is covered outside, the heat is not lost but retained. More protection should be given under a lug, at least two courses of brick and an air space be open above the lug.

Reverting to strength, note that it is usual to have a space of 4 inches between the boiler and the side wall, and by properly carrying the brick out to the boiler the weight is transmitted over the entire seat of the lug. On the other hand, if this is not done, then one must make the lug stronger to allow for the 4-inch span. When of cast iron, we cannot, as stated, accept more than 15,000 pounds tensile strength per square inch, while if of steel we can take 60,000 tensile strength as the ultimate strength. Hence a ¼-inch steel equals a 1-inch casting; but the lugs are made, if of steel, of equal width with ordinary cast-iron lugs, and in addition the pressed ribs make it much wider. It is usual

to have such lugs of $\frac{3}{8}$-inch plate, equaling castings of $1\frac{1}{2}$ inches in strength. Steel lugs are easily punched and fitted to boiler shells. There is life in them, as before reaching a breaking point through overload the give would be noticeable, while the casting would fail without warning.

From all of the above you will doubtless agree that in the modern steam boiler steel is winning its fight over cast iron through its superiority in strength and its adaptability for these purposes. In the up-to-date shops these arguments plus actual reduction in costs have led to its adoption.

XII

CARE OF THE HORIZONTAL TUBULAR BOILER[1]

ALTHOUGH the boiler room is the very heart of every steam plant, it is frequently the subject of the grossest neglect, and the instances in which it receives the care and thought to which it is entitled are very rare. Under the very best of conditions it is wasteful, but in a great many, in fact in the majority of cases, it is much more so than there is any necessity of. It must be admitted that the boiler room is necessarily a rather dirty and uninviting place, but that is no excuse for neglecting it.

In the engine room every possible economy is practised. Every foot of steam pipe is covered; the best grade of oil is used; the engine valves are set with the greatest care; belts are run as slack as possible; and many other points are watched in order to keep the steam consumption down to the lowest possible point. This is all very proper and good, and should be encouraged as much as possible, but in many cases far more serious losses are permitted in the boiler room, and it is these which can and should be stopped.

[1] Contributed to Power by M. Kennett.

A Common Source of Loss

A very common source of loss is the leakage of cold air through cracks in the settings. When flat plates from one side wall to the other are used over the rear of the combustion chamber, it is not unusual to find a space of from $\frac{1}{2}$ to 1 inch between the rear boiler head and the plate. This admits a large quantity of cold air to pass directly through the upper tubes, which are the most valuable for generating steam.

As a rule these openings are not caused by faulty setting of the plate, as these are usually well set, making a tight joint, before the boiler is fired up. But when the boiler becomes heated and expands, the plate is forced back, and when the boiler cools, a small space is left between it and the plate. Pieces of mortar and chips of brick lodge here, and when the boiler is again fired up and expands, the plate is forced still farther back. It is practically impossible to prevent these openings with this style of plate, but matters can be greatly improved by packing the crack loosely with waste which has been filled with soft fire-clay, as this forms an elastic packing which will not readily burn out.

The style of arch shown in Fig. 74 is practically free from this objection as it rests against the boiler head and follows its movements. This plan is open to the objection that the angle iron on the boiler head finally burns out, and in order to replace it the studs have to be removed from the head, and new ones put in, with the attendant trouble of making the job tight. Bear-

ing bars are sometimes built into the side walls as a substitute for these angle irons, but they soon burn out, also. All trouble from this source may be overcome by using an extra heavy pipe as a bearing bar, and making it part of the feed line, so that water is being constantly pumped through it. Or, as in one case in mind, a small open tank may be provided for this purpose, the water circulating by gravity.

Numerous other cracks are constantly developing in various parts of the settings, and should be kept well filled with clay.

The combustion chamber back of the bridgewall should not be allowed to become filled with ashes to such an extent as to impede the draft.

Proper Depth of Combustion Chamber

There is a great difference of opinion concerning the proper depth of this chamber, some engineers claiming that it should be filled up level with the bridgewall, while others claim it should be quite deep. Much depends upon the kind of fuel used, the writer believing that when soft coal is used, it should be fairly deep to allow the unburned gases to become thoroughly mixed with the air and burn. An excellent plan is to slope the flame bed from almost the hight of the bridgewall to the ground in the rear, and pave it with firebrick. This brick paving becomes incandescent and ignites the gases and also reflects the heat upward toward the boiler. This style of chamber is easily cleaned out through a door in the rear wall at the level of the floor, and the ashes should be raked out every day.

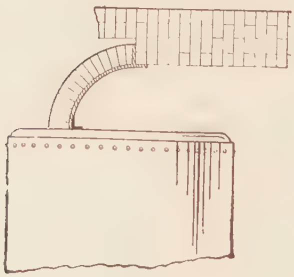

FIG. 74.

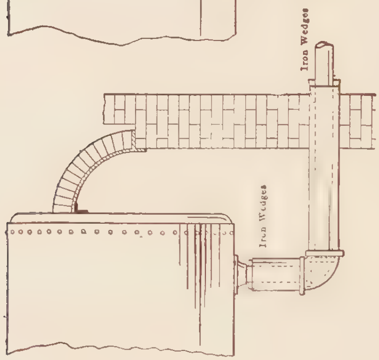

FIG. 75.

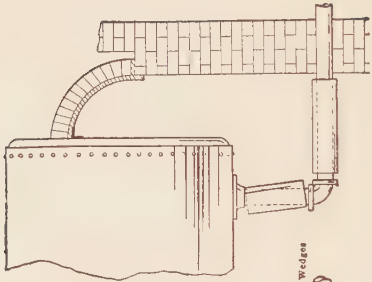

FIG. 76.

Do not make the mistake of placing this door one or two feet above the ground, as is frequently done, thus making it necessary to enter the chamber to clean it.

The hight and form of the bridgewall are also worthy of consideration. The principal object of the bridgewall is to keep the fire on the grates, and it should not be built up too close to the boiler, nor should it be curved to conform to the circle of the boiler shell. Such walls tend to concentrate the intense heat in the fire-box. This burns out the fire-door linings, and increases the danger of burning the fire-sheet, in case scale or sediment collects on it. They also prevent the free passage of a sufficient quantity of air into the combustion chamber to burn the gases.

We believe that all bridgewalls should be straight and not closer to the shell than 12 inches for 48-inch boilers, and 20 inches for 72-inch boilers. In many cases we believe these distances may be increased with advantage.

The necessity of keeping the tubes free of soot is pretty well understood, and this point usually receives proper attention. In addition to the usual blowing out with the steam blower, however, they should be thoroughly scraped at least once a week. The steam from the blower condenses to a great extent before reaching the rear end of the tubes; a great deal of the soot is simply moistened and left adhering to the tubes and soon burns into a hard scale which can only be removed by a thorough scraping.

Importance of Cleaning Boilers

Generally speaking, there are few operations about a steam plant which are so badly neglected as the cleaning of the boilers. The operation too often consists simply of letting the water out, removing the lower man-head, and washing the mud out with a hose. The natural result is that the heating surfaces, especially the tubes, become heavily coated with scale. This accumulates most rapidly at the rear head, and the space between the tubes soon becomes entirely choked for a short distance, preventing the free access of the water to the tube-sheet. This causes the tube ends to become overheated and they begin to leak. The only remedy is to remove the scale and reroll the tubes, but in order to remove the scale it is usually, or at least frequently, necessary to cut out some of the tubes.

The bagging of boilers due to the accumulation of scale and dirt is of such common occurrence as to require no discussion, other than to say that it is the result of improper cleaning.

Of course there are many cases in which, even with the best possible care, a great deal of scale will form, or where it is impossible to keep the boiler out of commission long enough to clean it properly. But there are also many cases in which a pretty thorough cleaning could be given if the engineer really wanted it done, and realized its importance sufficiently to see that it was done.

The number of boiler compounds which are guar-

anteed to keep boilers clean is legion, but still it will be hard to find the one which will remove the scale and hand it to the engineer, although this seems to be what some men expect of it. Most of them will do all that can be expected of them. They will soften and loosen the scale and considerable of it will drop off. After it is loosened the boiler cleaner should scrape it off by entering the top and bottom with suitable tools. Boiler compounds, like many other things, should be mixed with a good deal of common sense, then results will be obtained. When a boiler is badly scaled great care must be exercised in the use of a scale solvent, as it may cause considerable scale to drop off and bag a sheet. The action can be watched by frequent cleanings and, if there seems to be danger of such trouble, more frequent cleaning may be resorted to, or less solvent may be used.

An excellent plan is to use a scale pan, which is a shallow pan about four to six feet long and as wide as can be passed through the manhole. It is supported by light legs about three inches long, and is placed on the fire-sheet directly over the grates. As the scale falls it is caught by this pan and is thus kept off the sheet, preventing the bagging of the latter.

Oil a Source of Difficulty

Probably the most difficult thing to cope with in a boiler is oil. There are many different kinds of oil. Genuine crude petroleum is oil, but when properly used, it is difficult to find anything which excels it for keeping boilers free from scale. Kerosene is frequently used

for the same purpose, and neither causes any trouble. The oil which we refer to, and which causes the most trouble, is the cylinder-oil carried over by the exhaust to an open heater or hot-well, and from thence into the boilers. This first appears at about the water line, and on the top tubes, where it gives no trouble, but it soon spreads over the entire heating surface, and it is surprising how little it takes to cause a very serious bulge on a fire-sheet.

A bulge caused by oil is different from one caused by scale or mud in that it usually covers considerable area, while the latter is not often over a foot or 18 inches in diameter, but is much deeper in proportion to its size. When a bulge is from three to five feet long, as those caused by oil usually are, there is nothing to be done but to put in a new sheet. This is an expensive repair, as it necessitates tearing down the brickwork in addition to the boiler work.

The best method of removing the oil from the feed-water is to filter it through a bed of coke and excelsior. This must be renewed from time to time, as it soon gets coated with the oil and becomes useless. There are numerous separators on the market guaranteed to extract the oil from the steam and water, but invariably better results have been obtained from the filter.

Faulty Blow-off Pipes

The records of a large boiler-insurance company show that there are more claims due to the failure of blow-off pipes than from any other single cause. A volume might be written on this, for the blow-off pipe

certainly is a very troublesome, although necessary evil. When the feed-water is not introduced through the blow-off pipe, there is practically no circulation in it, and mud and sediment are very apt to collect. If the pipe is not protected from the direct action of the fire, this is very liable to cause it to burn and burst. Even though this results in no damage, it necessitates cutting out the boiler, and this may happen at a very inopportune time. If, however, the boiler is fed through the blow-off, the danger of such accident is reduced to the minimum, but even then it is better to protect the pipe from the direct action of the flame and gases.

It is a very common practice to incase the pipe in a sleeve formed of a pipe one or two sizes larger. This is of no value unless the sleeve is arranged as in Fig. 75, so as to allow a circulation of air between it and the blow-off pipe. When the sleeve is simply slipped over the pipe and allowed to hang loose, as in Fig. 76, it is of no value whatever.

In order to make it effective, the sleeve should inclose the entire pipe from the outside of the setting to within an inch or so from the boiler, and should be held in position by iron wedges, as shown in Fig. 75. This allows the cool air to traverse the entire pipe, being drawn in by the draft. While this is an excellent plan theoretically, it is open to the very serious objection that the sleeve rapidly burns out, and in order to renew it, the entire blow-off pipe has to be taken down. There is a cast-iron split sleeve made for this purpose, which can be replaced at any time without disturbing the pipe.

Perhaps as good a plan as any, all things considered, is to run the blow-off pipe straight down to the bottom of the combustion chamber and build a V-shaped fire-brick pier in front of it, just far enough away to allow removing the pipe and replacing it without disturbing the pier.

When necessary to use fittings in the combustion chamber, they should be of either cast steel or malleable iron, as cast iron is too liable to crack when exposed to high temperature.

Best Method of Feeding a Boiler

The method of feeding boilers has had a great deal of discussion, some advocating feeding through the blow-off, and some as strongly advising the top feed with a certain type of heater, the water passing through a length of pipe in the steam space of the boiler, and thus becoming heated to more nearly the temperature of the water in the boiler before discharging.

It is the general opinion the top feed is, generally speaking, the proper method; but circumstances must necessarily determine the best method for each particular case, and the writer has seen many cases where he has advised feeding through the blow-off. The objection to this method is that the comparatively cool water from the heater is discharged on the hot sheets. The water from the heater is hot, it is true, but when compared to that in the boiler there is considerable difference in temperature. It is seldom that the ordinary exhaust heater, except the most modern open heaters, raises the water to more than 175 degrees,

while the temperature of the water in the boiler at 100 pounds pressure is 337 degrees. This is a difference of 162 degrees, or about the same difference as between boiling water and a block of ice. Now if this water is passed through twelve or fourteen feet of pipe in the steam space before it is discharged, its temperature will be raised, perhaps not very much, but at the end of this pipe it is discharged in the body of water in the boiler, and cannot come in contact with the sheets until it has mingled with and attained the temperature of this water. If an injection is used, or if there is no heater used in connection with the feed-pump, the top feed should be used by all means.

It is fully realized that with some waters this internal pipe soon chokes up, especially at the end, but usually this is readily cleaned, when the boiler is cleaned, and it may easily be made of sufficient area to run three or four weeks without giving any trouble.

The point of discharge for this pipe is largely a matter of personal preference, but it should be remembered that the sediment will collect worst at the point of discharge. A good plan is to have the pipe enter the front head just above the tubes and at one side of the boiler, carrying it to within two or three feet of the back head, and supporting it by brackets from the braces. From here let it run across to the middle space between the tubes, using a union near the end of this piece. Then drop two pipes between these tubes to the level of the lower tubes. This makes it very easy to clean these down pipes, by opening the union and removing them.

If there is no manhole below the tubes, so that the scale and sediment cannot be scraped from the shell and tubes at this point, it may be found better to discharge at the side near the rear and just below the water line.

If for any reason the blow-off pipe cannot be arranged so that it can be properly protected from the fire (and occasionally this is the case), and if a good heater is used, there is no great objection to feeding through the blow-off if that is the preference of the engineer. It is always well to have both systems installed so that if one fails the other may be used.

XIII

CARE AND MANAGEMENT OF BOILERS [1]

THERE has been a great deal written by different authors on the subject of care and management of boilers. Valuable advice has been given, yet boiler explosions and accidents still occur. Therefore, too much cannot be said to impress upon the mind of the stationary engineer the importance of taking care of boilers.

The first and most important thing to begin with is a good, sound boiler, for if the boiler is an old and dilapidated concern the best and most skilful engineer cannot make it safe and reliable, and the only advice in any case like this would be to have nothing to do with it, as not only his reputation as an engineer would be at stake but also his life and the lives of others.

When taking charge of a plant that has been run for some time the engineer should lose no time in ascertaining as far as possible the exact condition of the boilers, and at the first opportunity he should make an internal and external examination and see that they are free from scale and incrustation. If they are not, he should see that they are thoroughly cleaned both inside and outside of the shell. When a boiler is once

[1] Contributed to Power by John McConnaughy.

thoroughly cleaned the competent engineer will always resort to the proper means of keeping it so, as far as conditions will allow.

The accumulation of scale can be in a measure avoided by blowing small quantities of water from the bottom and surface blow-off, as all minerals held in suspension become of greater specific gravity than the water. When heated, the tendency by specific gravity is to settle toward the bottom while the lighter portions remain upon the top and float in the form of a scum. It has been found that by frequently blowing from the surface and bottom blow-off, much of the mineral substance which forms scale will be carried out before it can settle sufficiently to attach itself to the iron. By so doing, much of the trouble from scale may be avoided.

Notwithstanding all the care that may be taken, in some localities where the water is largely impregnated with minerals a certain amount of scale will accumulate in spite of the efforts of the most careful and experienced engineer. There are various devices and compounds on the market which have proved effective and in a measure beneficial for preventing this scale. Others are of a doubtful character; it is advisable before using a compound to have a chemical analysis made of the feed-water, as the nature of the supply receives too little attention.

Some engineers having charge of boilers with manholes under the tubes do all their cleaning from below the tubes and do not open the boiler on top. As it is impossible to wash all the dirt down from the

top by washing from the under side of the tubes, the boiler is in bad condition above the tubes before they know it and they will tell you that the boilers are in good shape inside.

In cleaning boilers, all manholes and hand-hole plates should be taken out and the washing should be done from above and below the tubes. The engineer should then go inside the boiler and clean between them, so that any scale that has been lodged between the tubes can be taken out. On the outside, all seam heads and tube ends should be examined for leaks, cracks, corrosions, pitting and grooving. The condition of stays, braces and their fastenings should be examined. The shell of the boiler should be thoroughly cleaned on the outside, as soot is a bad conductor of heat, holds dampness and is liable to cause corrosion. All valves about the boiler should be kept clean and in good working condition. The pumps or injectors should be in the best working order. The connections between the boiler and water column and also the gage glass should receive the closest attention, but they are sadly neglected by some engineers. The brickwork should be kept in good condition and all air holes stopped, as they decrease the efficiency of the boiler and are liable to cause injury to the plates by burning.

There should be a good heater in connection with the boiler and the feed-water as hot as you can work it, for feeding cold water causes too much contraction and expansion. This causes vibration in the seams and makes them weak at those points. For example, if one hundred pounds of steam will do your work; never

carry any more nor less, as the rise and fall in pressure causes expansion and contraction of the plates.

Never open the fire doors to cool your boiler. Close the ash-pit doors and open the smoke-box doors in case you get too much steam, as opening the fire door causes too much contraction by the cold air cooling the furnace. It would be better to allow steam to blow off from the safety valve, which will not in any way injure the boiler.

The safety valve should be raised from its seat every day to make sure it does not stick from any cause, and observe from the steam gage if the valve blows off at the pressure it is set for.

It is of the highest importance to keep the blow-off pipe free from sediment of any kind, as the pipe is liable to fill up and burn off, and the only way to keep it free is to open the blow cock often enough to keep everything flushed out.

The best time to blow off is in the morning before the fires have been started up, as a good deal of sediment in the boiler will then have settled to the bottom of the shell and much of it will pass out when the cock is opened. Noon is also a good time, after the fires have been banked for half an hour or more, so that the water in the boiler has been quiet long enough to deposit the particles that are being whirled about with it through all parts of the boiler.

When a blow-off cock is opened, it must be remembered that it is not to be yanked wide open and then closed the same way. This practice is very dangerous. No valve about a steam system ought to be closed

suddenly, except in time of emergency, because the sudden strain on the pipe and fittings is liable to cause a rupture in the pipe or else break the elbow or valve.

The boiler is the life of any plant and my advice to all owners of steam plants is to keep a first-class engineer, one who is strictly temperate, pay him good wages, give him the necessary material, and his plant will get the proper care and management.

XIV

SETTING RETURN TUBULAR BOILERS

A GREAT improvement is made when we discard the old-time setting of return-tubular boilers, in which cast-iron brackets were supported by brick walls which are constantly crumbling away, for the substantial form of setting which is obtained by suspending return-tubular boilers from I-beams supported by cast-iron columns.

The accompanying Figures 77 and 78 show the setting of boilers in single or double batteries. In setting an even number of boilers, as six or eight in one setting, it is best to divide them into pairs so that not more than two boilers will be suspended between supports.

The principal reason for this is that when the large sizes, such as from 150 to 250 horse-power are used, the size I-beam required to safely carry this load between supports is so large that it overbalances the cost of two or more cast-iron columns.

In setting an odd number of boilers, such as three or five, in a battery, columns are usually placed between each boiler with a 2-inch air space all around the column and an air duct at the bottom of the setting which runs through from the front to the back and connects with

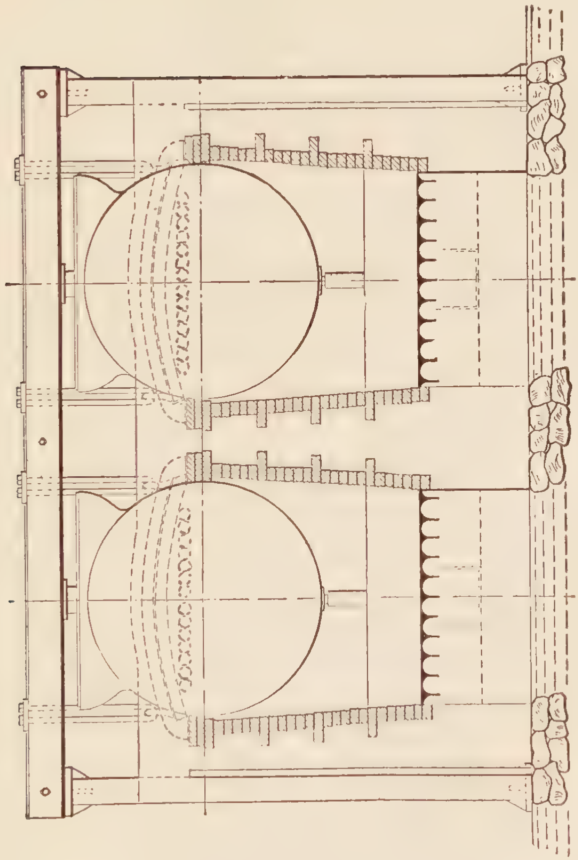

FIG. 77.

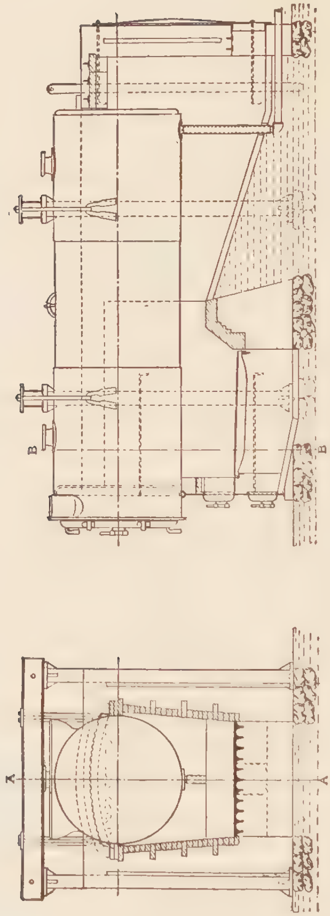

Fig. 78.

each air space around the column. This keeps up a circulation of air and the columns are kept comparatively cool.

In setting boilers in this manner the columns and I-beams are set in position first. Then the boiler is hoisted to the proper hight by means of tackle which is fastened to the I-beams and when the boiler is brought to the proper hight the U-bolts are slipped into place and fastened by nuts and washers to the I-beams. This does away with the use of blocking and barrels which are generally used and leaves all the space clear under the boilers.

The expansion is easily taken care of by the U-bolts and hangers, as is shown in the setting plans, and if the walls are properly set, they should show no cracks as they carry no weight and are entirely free.

The accompanying table has been carefully worked out with a factor of safety of 5 and gives the different lengths and sizes of I-beams and columns required, so that a person estimating on a job of this kind can readily determine the cost of such a setting. It covers boilers from 36 inches in diameter and 8 feet long to 90 inches and 20 feet, giving the total weight to be supported and the sizes, weights and positions of columns and I-beams required.

SIZES AND WEIGHTS OF COLUMNS AND I-BEAMS RE

HORSE POWER	15	20	25	30	35	40	45	50	60
Dia. of boiler in inches	36	36	42	42	44	48	50	54	54
Length of tubes in feet......	8	10	10	12	12	12	13	13	15
Length of curtain sheet in inches..................	11	11	12	12	12	14	14	14	14
Total weight of boiler and water	9500	7500	9400	10500	11500	13300	14200	15300	17800
Rear head to center of hanger	2–0	2–6	2–6	3–0	3–0	3–0	3–3	3–3	3–6
Center to center of hangers .	4–0	5–0	5–0	6–0	6–0	6–0	6–6	6–6	7–0
Front head to center of hanger	2–0	2–6	2–6	3–0	3–0	3–0	3–3	3–3	3–6
Distance between C of supports (1 boiler)	6–0	6–6	7–0	7–0	7–2	7–6	7–8	8–0	8–0
Distance between C of supports (2 boilers)	11–8	11–8	12–8	12–8	13–0	13–8	14–0	14–0	14–8
Length of I-beam for 1 boiler.	7–3	7–3	7–10	7–10	8–0	8–4	8–6	8–10	8–10
Length of I-beam for 2 boilers.	12–6	12–6	13–8	13–8	14–0	14–8	15–0	15–10	15–10
Size of I-beam required for one boiler	4	4	5	5	5	6	6	6	6
Size of I-beam required for 2 boilers	6	6	8	8	9	9	9	10	10
Weight per ft. of I-beam for one boiler	7.5	7.5	9.75	9.75	9.75	12.25	12.25	12.25	12.25
Weight per ft. of I-beam for two boilers	12.25	12.25	18	18	21	21	21	25	25
Length of cast-iron column ..	8–0	8–0	8–6	8–6	8–8	9–3	9–5	10–0	10–0
Outside Dia. of C. I. col. for one boiler	4	4	4	4	4	5	5	5	5
Outside Dia. of C. I. col. for two boilers	5	5	5	5	5	6	6	6	6
Size of flange on ends of col. for one boiler	9½	9½	10	10	10	10½	10½	10½	10½
Size of flange on ends of col. for two boilers	10½	10½	12	12	12½	12½	12½	13½	13½
Thickness of C. I. col. for one boiler	½	½	½	½	½	½	½	½	½
Thickness of C. I. col. for two boilers	¾	¾	¾	¾	¾	¾	¾	¾	¾

QUIRED IN SETTING RETURN TUBULAR BOILERS.

70	75	80	90	100	125	150	175	200	200	225	225	250
60	60	60	66	66	72	72	78	78	84	84	90	90
14	15	16	15	16	16	18	18	20	18	20	18	20
16	16	16	17	17	18	18	18	18	20	20	22	22
20800	24800	27200	30300	35000	40000	44000	48000	56000	55000	67000	65000	75000
3–6	3–9	4–0	3–9	4–0	4–0	4–6	4–6	5–0	4–6	5–0	4–6	5–0
7–0	7–6	8–0	7–6	8–0	8–0	9–0	9–0	10–0	9–0	10–0	9–0	10–0
3–6	3–9	4–0	3–9	4–0	4–0	4–6	4–6	5–0	4–6	5–0	4–6	5–0
9–0	9–0	9–0	9–6	9–6	10–0	10–0	10–6	10–6	11–0	11–0	11–6	11–6
16–2	16–2	16–2	17–2	17–2	18–2	18–2	19–2	19–2	20–2	20–2	21–2	21–2
10–0	10–0	10–0	10–6	10–6	11–0	11–0	11–7	11–7	12–0	12–0	12–6	12–8
17–4	17–4	17–4	18–4	18–4	19–5	19–5	20–6	20–6	21–6	21–6	22–6	22–6
7	7	7	7	7	8	8	9	9	9	9	9	10
12	12	12	12	12	15	15	15	15	15	15	15	15
15	15	15	15	15	18	18	21	21	21	21	21	25
31.5	31.5	31.5	40	40	42	42	60	60	60	80	80	80
10–8	10–8	10–8	11–2	11–2	12–0	12–0	12–6	12–6	13–0	13–0	13–10	13–10
5	5	5	6	6	6	6	6	6	6	6	6	6
6	6	6	6	6	8	8	8	8	8	8	8	8
11½	11½	11½	11½	11½	12	12	12½	12½	12½	12½	12½	13½
14	14	14	14½	14½	15	15	16	16	16	17	17	17
¾	¾	¾	¾	¾	¾	¾	1	1	1	1	1	1
¾	¾	¾	1	1	¾	¾	¾	¾	1	1	1	

XV

RENEWING TUBES IN A TUBULAR BOILER[1]

WHILE the renewal of boiler tubes is properly the work of the boiler maker, the engineer who knows how to and can do it is just so much more valuable to the employer. The purpose of this article is to describe the method employed, together with the tools required.

First, it is essential to place a distinguishing mark on the front and rear heads to show which tube is to be cut out, using chalk or soapstone for the purpose, and the best way to make sure that the helper at the other end of the boiler marks the same tube that you do is to run through a strip of wood four or five inches longer than the tube. As such a strip is of use farther along in the process it is well to make a length of $\frac{7}{8} \times 2$-inch pine to serve both purposes. Next, with a hammer and a heavy cape chisel having a wide cutting edge, which is less liable to cut or mar the boiler (see Fig. 79), face the beads on both ends of the old tube until they are flush with the heads of the boiler. Then, at the front head, with a diamond-point chisel such as is shown in Fig. 80, cut a slot or channel, $\frac{1}{16}$ inch wide, in the

[1] Contributed to Power by J. E. Sexton.

bottom of the tube, extending inward to about $\frac{3}{8}$ of an inch beyond the inner edge of the head, making sure that the groove is cut in the tube only and that the head

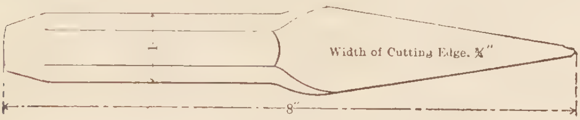

Fig. 79.

is not cut or even marked by the chisel. *Do not drive the chisel clear through the tube, either.*

With an offset chisel, Fig. 81, carefully turn up the

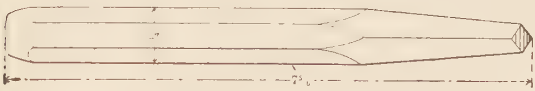

Fig. 80.

edges of the tube at both sides of the cut, until the tube-end resembles the condition shown in Fig. 82, when it will be found that this end of the tube has

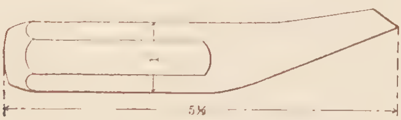

Fig. 81.

been released from the head. In cutting the slot, especially after the cutting edge of the chisel has gone beyond the thickness of the head, if the chisel is allowed to go through the tube it will be the source of

considerable trouble, as it will cause the tube to spread. Hence, at this point extreme care must be used.

If a tube is corroded and muddy, it will be harder to remove and the method will have to be changed somewhat. Considerable force is required sometimes to remove such a tube. Instead of one slot in the bottom of the tube, two are cut, about $\frac{3}{8}$ of an inch apart, and

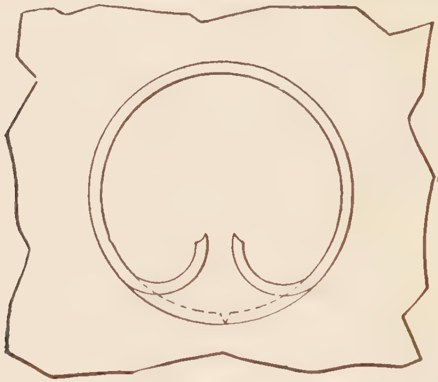

FIG. 82.

the offset chisel is used as before, except that the $\frac{3}{8}$-inch piece is turned up until it looks like the letter C, with its back toward the front of the boiler. Then proceed as before, turning the edges of the cut upward as far as they will go. A hook on the end of a chain or rope may then be inserted in the loop formed by the C-piece. This takes care of the front end.

At the other end of the tube insert the end of a piece of shafting about 10 inches long and a little smaller in diameter than the outside diameter of the tube.

The end of this shafting should be turned so that it will enter the tube about one inch, with an easy fit, and by giving a few taps on the outer end of this improvised mandrel the tube will be loosened at this end. Then, by working the tube backward and forward it can be released altogether.

The next step is to mark the new tube so it can be cut to length. Insert the $\frac{7}{8}$x2-inch piece of pine into the holes the old tube came out of until one end of the strip extends through the rear head about $\frac{5}{32}$ of an inch. Hold it there and proceed to place a mark on the end extending from the front head $\frac{5}{32}$ of an inch from face of the head. This gives the proper length to which to cut the new tube. Then, while the tube is being cut to length, take a half-round second-cut file, or a finish-cut file, and carefully smooth up the heads around the holes, removing any marks or cuts which may have been made in taking out the old tube. This is to prevent future leaks. Next, push the new tubes into place and station the helper at the rear end with a tube expander, being sure that the ends of the tube are equidistant from the heads. It is advisable to insert one end of an 8-foot section of 1-inch pipe in the front end of the tube, for a distance of 12 inches or so, and exert a downward pressure on the lever so provided to prevent the tube from turning while the rear end is being expanded. As soon as the tube is tight at the rear end, proceed to expand the front end.

A self-feeding expander, Fig. 83, will give good results, especially if a ratchet wrench is used to turn the spindle, for one can tell by the feeling just when

to stop expanding. A monkey wrench will do, however, if a ratchet wrench is not available.

The beading comes next. This requires a special tool similar to that shown in Fig. 84. Place the long prong of the tool inside the tube, with the short prong pressing against the tube-end. Then bead the tube-end

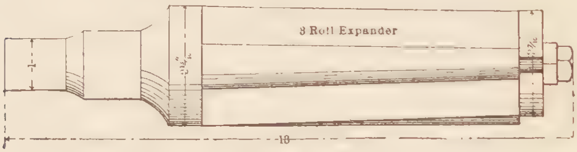

Fig. 83.

thoroughly *throughout the circumference*, for if it is only beaded here and there it will prove very unsatisfactory. When both ends are beaded, use the expander lightly in each end once more, to remove the marks made by the beader.

Fig. 84.

If both hand-hole plates are tight and the blow-off valve works O. K., fill the boiler with either hot or cold water, until the tube is covered, and if the tube does not leak water it will hold steam, and the boiler is ready to put into commission. If the tube leaks, re-expand it very lightly. Ordinarily, a man and helper can renew a tube in an hour, with ease.

XVI

USE OF WOOD AS FUEL FOR STEAM BOILERS[1]

In nearly all plants where lumber or wooden articles are the finished products, wood is used as a fuel for the boilers, because it is a refuse and is easily and cheaply disposed of in that manner. In some plants the amount of this refuse is greater than can be burned under the boilers; in others, there is not enough waste to furnish the steam required.

This is the case in a great many wood-working industries, and in some sawmills on the South Atlantic coast. To this class of industries this article is directed.

A certain wood is a good fuel or a poor fuel, depending on (1) the moisture contained and (2) the size of the pieces as fired. Whether it burns well under the boiler depends on the shape of the furnace, the method of firing and the draft of the chimney.

Calorific Value of Various Woods

The main idea to be shown in this section is that the value of all woods is about the same, depending on the amount of moisture contained.

[1] Contributed to Power by J. A. Johnston.

In various works of reference, the weight of one cord of different woods (thoroughly air-dried) is about as follows, the quality of coal not being given:

Hickory or hard maple — 4500 lb. equals 1800 lb. of coal (others, 2000 lb.).

White oak — 3850 lb. equals 1540 lb. of coal (others, 1715 lb.).

Beech, red and black oak — 3250 lb. equals 1300 lb. of coal (others, 1450 lb.).

Poplar, chestnut and elm — 2350 lb. equals 940 lb. of coal (others, 1050 lb.).

Average pine — 2000 lb. equals 800 lb. of coal (others, 925 lb.).

Referring to the figures last given in each case in connection with "others," it is said:

"From the above it is safe to assume that $2\frac{1}{4}$ pounds of dry wood are equal to 1 pound of average quality soft coal, and that the fuel value of different woods is very nearly the same, that is, a pound of hickory is about equal to a pound of pine, assuming both to be dry."

It is important that the woods be dry in the comparison, as each 10 per cent. of water or moisture in the wood will detract about 12 per cent. from its fuel value.

Take an average wood of the chemical analysis: Carbon, 51 per cent.; hydrogen, 6.5 per cent.; oxygen, 42 per cent.; ash, 0.5 per cent. If perfectly dry, its fuel value per pound, according to Dulong's formula,

$$V = \left[14{,}500C + 62{,}000\left(H - \frac{O}{8}\right)\right]$$

is 8170 B.t.u. The calorific value of carbon equals 14,500 B.t.u., and the calorific value of hydrogen equals 62,000 B.t.u.

The hydrogen in the fuel being partly in combination with the oxygen, only that part not in such combination can be counted on as a fuel, hence the factor

$$\left(H - \frac{O}{8}\right).$$

If this wood as ordinarily dried in air contains 25 per cent. moisture, then the heating value of a pound of such wood is $8170 \times 0.75 = 6127$ B.t.u., less the heat required to raise the $\frac{1}{4}$ pound of water from atmospheric temperature to steam, and to heat this steam to chimney temperature. Say, for instance, it takes 150 B.t.u. to heat the water to 212 degrees and 966 B.t.u. to evaporate it to steam, and 100 B.t.u. to raise the temperature of the steam to chimney temperature; in all 1216 B.t.u. per pound or 304 B.t.u. per $\frac{1}{4}$ pound. The net value of the wood as a fuel would then be $6127 - 304 = 5824$ B.t.u., or about 0.4 that of 1 pound of carbon. This method can be applied to any wood, knowing its chemical analysis and its percentage of moisture as burned.

The Moisture Content

As nearly all woods have about the same chemical analysis, the heat value of woods depends, as before mentioned, almost entirely on the moisture contained in the wood when burned. When newly felled wood contains a proportion of moisture which varies much

in different kinds and different specimens, ranging between 30 and 50 per cent., and averaging about 40 per cent. Perfectly dry wood contains about 50 per cent. of carbon, the remainder consisting almost entirely of hydrogen and oxygen in the proportion which forms water. The coniferous (pines) family contains a small quantity of turpentine, which is a hydrocarbon. The proportion of ash in wood is from 1 to 5 per cent. The total heat of combustion in all woods is almost exactly the same, and is that due to the 50-per cent. carbon.

American woods vary in percentage of ash from 0.3 to 1.2 per cent., and the heat value ranges from 6000 B.t.u. for white oak to 9883 for long-leaf pine, the fuel value of 0.5 pound of carbon being 7272 B.t.u.

In the absence of any method of determining the heating value of a certain wood, the following are averages of the analyses of beech, oak, birch, poplar, and willow:

Carbon, per cent	49.70
Hydrogen, per cent	6.06
Oxygen, per cent	41.30
Nitrogen, per cent	1.05
Ash, per cent	1.80

These can be used in the foregoing formula, and will give an approximate value for nearly all American woods.

A very good and fairly accurate approximation of the amount of moisture in any particular sample can be obtained by weighing the wood (say about 10 pounds of it), and then placing the sample in a closed vessel

with a small hole in it to allow the steam to escape. Subjecting the whole to a temperature of about 220 degrees until all the moisture has been driven off, weigh the sample again, and the percentage of moisture in the original can be computed easily. With this percentage known, the subtraction for moisture present can be made, as before shown, and an approximate value of the sample is obtained.

Nearly all woods will give a heat value, dry, of about 8200 B.t.u. Having obtained the percentage of moisture present, the heat value of the fuel is 8200 multiplied by (100 per cent. — per cent. moisture) less (heat required to raise water contained to evaporative point) less (heat required to evaporate water) less (heat required to heat steam made by this water to chimney-gas temperature). All of the latter quantities can be obtained from steam tables.

Easiest Method for Getting at the Heat Value

Probably the easiest and most accurate of all methods of obtaining the heat value of a certain specimen of wood is not to inquire into the chemical analysis, but to take a sample of the wood just in the condition in which it is burned, place it in a closed, air-tight vessel, and keep it there until it is brought to a calorimeter. This instrument should be used by one who is familiar with its use. It will give the heat value expressed as B.t.u. per pound, dry. The percentage of moisture being found, the correction for moisture is made as before.

A case of this kind, taken from a report by the writer, may be mentioned and calculated. The fuel was sweet gum refuse from a veneer mill, run through a hog and ground into chips approximately the size of a man's little finger. The logs were brought to the mill by rafting down a river, so the chips as fed to the boilers were not out of the water over three-quarters of an hour. A sample of chips was weighed wet, then dried in a closed vessel and weighed again, giving a moisture percentage of 47.50. A sample of the dried wood was then ground and tested in a calorimeter, giving a heat value of 8208 B.t.u. per pound, dry.

For every pound of the wood fired there was only $8208 \times 0.525 = 4309$ B.t.u. given up by the wood in burning, for there was but $1.00 - 0.475 = 0.525$ pound of dry wood fired for 1 pound of fuel.

One pound of water requires 966 B.t.u. to evaporate it at the pressure in the furnace. There was 0.475 pound of water in the 1 pound of fuel fired so that $996 \times 0.475 = 458.88$ B.t.u. were required.

The flue-gas temperature was 340 degrees and $340 - 212 = 128$ degrees F. through which the moisture had to be raised as superheated steam. The specific heat of superheated steam at atmospheric pressure is about 0.48, and therefore the heat required for superheating the moisture in the fuel at the pressure of the furnace gases was $128 \times 0.48 = 61.44$ B.t.u. per pound of water in the fuel, or as $47\frac{1}{2}$ per cent of the weight of the fuel consisted of water, then this loss per pound of water in the fuel was 61.44×47.5 per cent $= 29.184$ B.t.u., so that the total loss of heat due to the presence of water in

the fuel was 458 + 29.184 = 487.184 B.t.u. per pound of the fuel, and therefore the available heat in the wood was about 4309 − 487 = 3822 B.t.u. per pound of the original fuel.

Probably the greatest chance of error in estimating the value of a wood as fired is to neglect the above calculation, because the difference between its heating value dry and its heating value as fired is often as high as 50 per cent., while a similar calculation for coal would give a comparatively small difference.

After computing by either of the methods given the heat value of the fuel to be burned, it is easily computed how much water can be evaporated per pound of fuel, and knowing the amount of available fuel, the power to be generated at any plant under consideration may be estimated.

Referring again to the same case of wet gum chips, this fuel was brought to the boiler house by a conveyer, the average capacity being measured at 100 pounds per minute, or 6000 pounds per hour brought to the boilers.

A pound of water requiring 966 B.t.u. to evaporate it, and each pound of the fuel having 3822 B.t.u., the evaporation from and at 212 degrees F. would be 3822 ÷ 966 = 3.95 pounds of water per 1 pound of fuel, and with 6000 pounds of fuel per hour, the maximum quantity of water that could be evaporated by the boilers, at 100 per cent efficiency, would be 6000 × 3.95 = 23,700 pounds of water on the basis of evaporation from and at 212 degrees F.

If the boilers were 70 per cent efficient, then 2370 ×

0.70 = 16,590 pounds of water per hour, evaporated from and at 212 degrees, is all that could be expected, and as 34½ pounds of water per hour evaporated from and at 212 degrees is equivalent of 1 boiler horse-power, the evaporation given would represent 16,590÷34.5 = 480.8 boiler horse-power. Under tests the boilers gave 450 boiler horse-power.

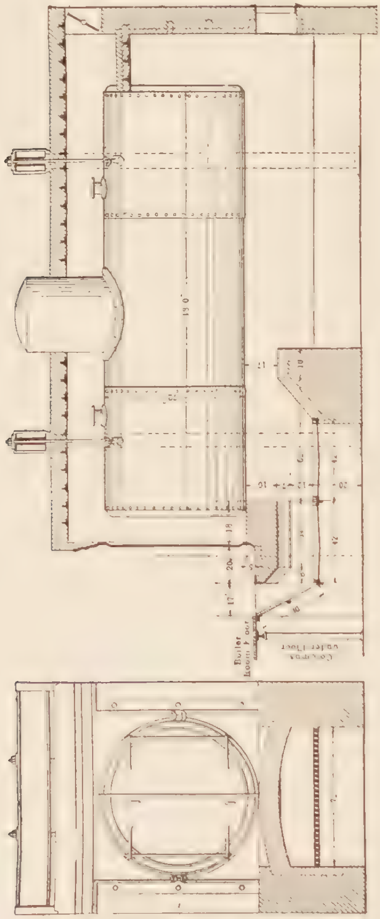

FIG. 85.

From all that goes before, it appears that wood as a fuel has been allowed a little too high a value, inasmuch as it is rarely if ever fed to the boilers perfectly dry. It is generally green, and in cases of sawmills located on the banks of navigable streams is soaked with water.

Air-drying of wood extracts about one-half of the moisture in a year. Wood perfectly dried, and then exposed to the air, will absorb about the same amount of moisture that it would contain after being thoroughly air-dried. However, when wood is to be used as a fuel, it is almost out of the question to contemplate drying it, so the proposition is to burn the fuel available in the best manner.

Fuel Available

Of course there cannot be given any even approximate method of calculating the amount of fuel that will be available in the refuse from any contemplated plant, for each and every one is to work under different conditions.

In plants already built, an estimate can be made by weighing the fuel brought to the boiler room, and by foregoing methods determining heat value, the available horse-power can be computed.

Most sawmills furnish enough refuse in slabs to run the boilers required to operate the plant. Woodworking plants, sash, blind and door manufactories, furniture factories, etc., depend entirely on the kind of product, as to the amount of scrap.

This will also depend largely on the plant at which the installation is contemplated. Furniture factories, woodworking plants, etc., generally work the kiln-dried lumber up so closely that the refuse as it comes to the boiler is already in an easily burnable condition, that of sawdust, shavings, or small strips or blocks. These can be fed directly to the furnace without further preparation.

In most sawmills where the slabs come off of the logs in long pieces, it is not possible to get the fuel to burn easily if fed as slabs, so it is often and generally in the sawmills on the Atlantic coast fed through a hog which grinds the slabs into chips varying in size from a man's three fingers to one finger or smaller.

This is undoubtedly the best way in which to introduce this fuel to the boiler, for it is then easily handled by conveyers, and can be dumped directly into the fire without any manual work, while slabs will generally require handling, unless some extra design is prepared to meet the case.[1]

The various forms in which wood is fed to the furnace may be summarized as: Cordwood, shavings, sawdust, dust from a hog, strips and blocks from a factory, and tan-bark.

Kind of Furnace Required

Efficient burning of wood requires a large combustion chamber, and grates arranged to prevent admission of a surplus of air. This cannot be obtained to good advantage in the usual coal-burning furnace, so the dutch oven has been developed to meet requirements. This is an extension of the fire-box in front of the boiler, as shown in Fig. 86, with a firing hole in the top through which the ground fuel or sawdust can be fed directly from the conveyer or chute to the grate.

As wood fuel is generally wet, or contains a large amount of moisture, the conditions of success, as pointed out by Thurston, are: To surround the mass so completely with heated surfaces and with burning

[1] A case of this kind is mentioned in Power, November, 1907.

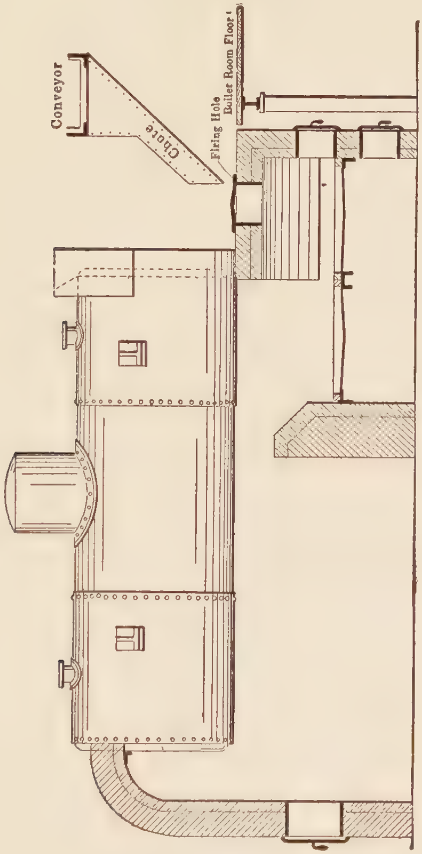

FIG. 86.

fuel that it may be rapidly dried, and so arranging the apparatus that thorough combustion may be then secured, the rapidity of combustion being precisely equal to, and never exceeding, the rapidity of drying. If the proper rate of combustion is exceeded, the dry portion is consumed completely, leaving an uncovered mass of fuel which refuses to take fire.

These conditions are met in the dutch oven, because of the fact that the fire is completely surrounded by fire-brick walls, which become heated to a very high temperature, especially in the case of burning pine shavings. This condition of good burning has been so well met in some cases that the fire-brick lining could not withstand the high temperature longer than a month.

The dutch oven has a firing door and an ash door on the front; the firing door may be used to fire any large pieces of wood, but results are best where the fire door on the front is never opened except for cleaning.

Fig. 87 shows an arrangement where the fuel consisted almost entirely of kiln-dried refuse from a woodworking plant, coming to the boilers in short sticks from about $\frac{1}{2} \times \frac{1}{2} \times 12$ inches to blocks $1 \times 3 \times 10$ inches, all mixed with sawdust and shavings from planers.

In the boiler room the floor is on an exact level with the top of the dutch oven. The fuel is dumped from a conveyer on this floor and shoved by hand into the holes on top of the ovens, and as the holes are kept full of fuel all the time, the doors over them are never closed. The boilers are of the Heine water-tube type,

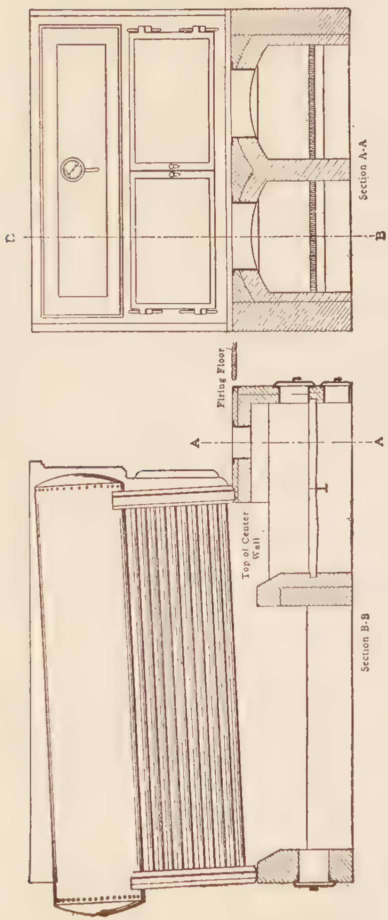

FIG. 87.

arranged in a battery of three and each rated at 300 horse-power. This installation gives perfect satisfaction.

Another form of combustion chamber, shown in Fig. 85, is very satisfactory for burning sawdust with a small mixture of shavings. The grate must be kept covered all the time, or too much air will get through, thereby decreasing the efficiency of the boiler. In this case the fuel is fed in a constant stream from a chute and is shoved back over the grate by a man on the firing floor.

For ordinary air-dried cordwood, a good grate is one placed at the firing-floor level, the area of grate being reduced to about two-fifths the amount required for coal by sloping the furnace walls inward, beginning just under the arch. The grate is, of course, at the bottom, and the cordwood can be carried to a depth of 30 to 36 inches, so that the freshly fired wood will crowd down that which is partly burned, filling the large interstices at the bottom with burning coals, and preventing leakage of air past the fire.

Miscellaneous Points

In handling any kind of wood fuel, it is better, even in small installations, to have the fuel brought by some carrier, as a conveyer, chute or air blast, to the furnace. With dry wood in small pieces, as dust from a hog, or shavings, the fuel being brought to the fire-room, one man can care for about 300 horse-power of boilers. If it is brought right over the firing hole to a dutch oven by an overhead carrier, he can care for, in some cases, 500 horse-power.

Sawdust and dry shavings are very extensively handled by blowers, the suction of the blower being connected to the saw frame or planer, and the refuse being blown into a receptacle over the boiler room. It is then dropped by chutes directly into the fire, or may be blown directly in by the blast furnishing air for the fire.

A chimney could be designed from theoretical calculations involving the chemical composition of the wood to be burned, but as a plant burning wood is rarely or practically never run on a weight basis, this would not be a practical method.

It has been borne out by practice that a chimney designed for a certain horse-power for bituminous coal will work well for wood. The accompanying curves were calculated from Kent's formula:

$$\text{H. P.} = 3\tfrac{1}{2}\,\text{E.}\sqrt{\text{H.}}$$

In Figs. 88 and 89, with any boiler horse-power and any suitable hight, the area of stack can be found. In Fig. 90 this area is expressed for round or square stacks.

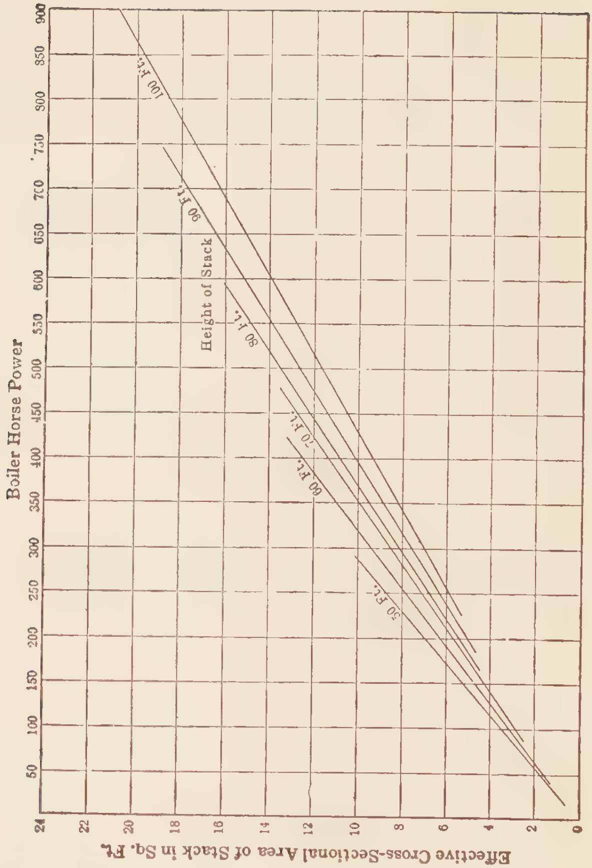
Boiler Horse Power
Effective Cross-Sectional Area of Stack in Sq. Ft.
Height of Stack
100 Ft.
90 Ft.
80 Ft.
70 Ft.
60 Ft.
50 Ft.
50
100
150
200
250
300
350
400
450
500
550
600
650
700
750
800
850
900
24
22
20
18
16
14
12
10
8
6
4
2
0

FIG. 88.

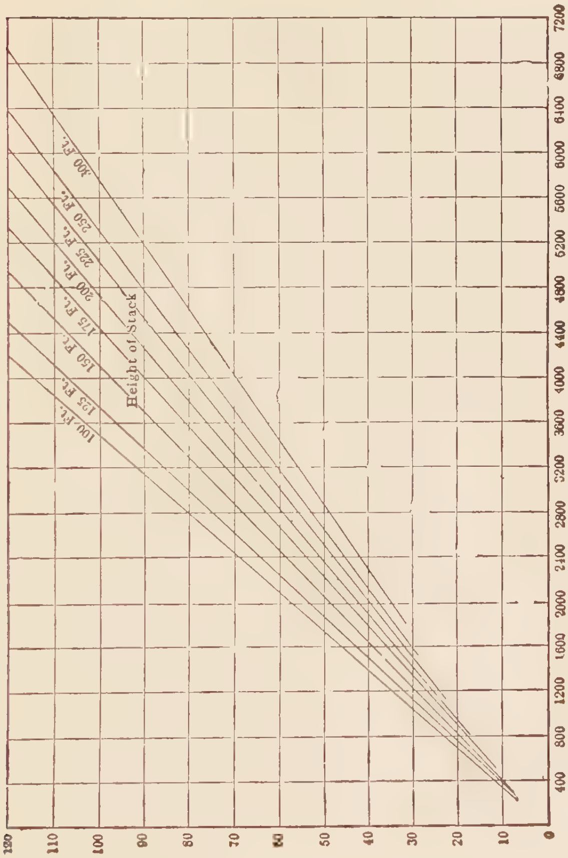
Height of Stack
300 Ft.
250 Ft.
225 Ft.
200 Ft.
175 Ft.
150 Ft.
125 Ft.
100 Ft.
7200
6800
6400
6000
5600
5200
4800
4400
4000
3600
3200
2800
2400
2000
1600
1200
800
400
120
110
100
90
80
70
60
50
40
30
20
10
0

Fig. 89.

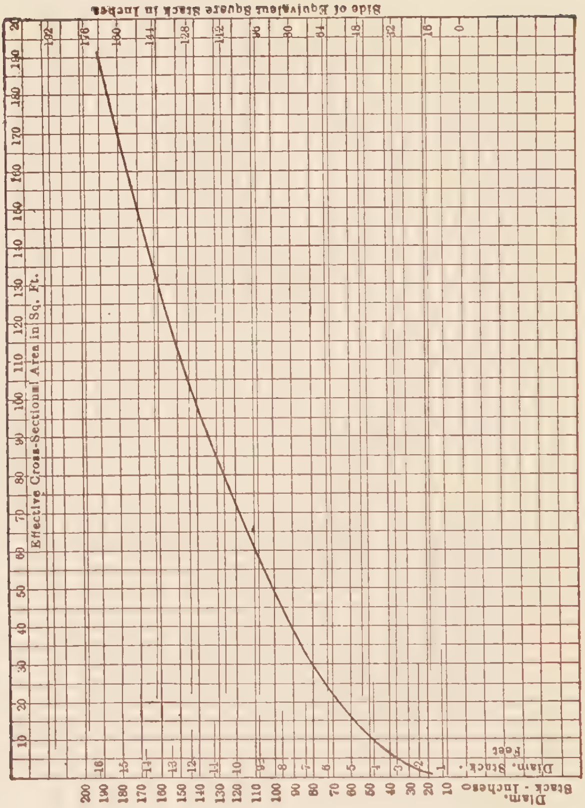
Side of Equivalent Square Stack in Inches
Effective Cross-Sectional Area in Sq. Ft.
Diam. Stack - Inches
Diam. Stack - Feet

FIG. 90.

XVII

BOILER RULES

THE Board of Boiler Rules appointed under a recent act of the Massachusetts legislature has adopted the following regulations:

SECTION 1 — MAXIMUM PRESSURE ON BOILERS

1. The maximum pressure allowed on any steam boiler constructed wholly of cast iron shall not be greater than twenty-five (25) pounds to the square inch.

2. The maximum pressure allowed on any steam boiler the tubes of which are secured to cast-iron headers shall not be greater than one hundred and sixty (160) pounds per square inch.

3. The maximum pressure allowed on any steam boiler constructed of iron or steel shells or drums shall be calculated from the inside diameter of the outside course, the percentage of strength of the longitudinal joint and the minimum thickness of the shell plates; the tensile strength of shell plates to be taken as fifty-five thousand pounds per square inch for steel and forty-five thousand pounds per square inch for iron, when the tensile strength is not known.

Shearing Strength of Rivets

4. The maximum shearing strength of rivets per square inch of cross-sectional area to be taken as follows:

Iron rivets in single shear	38,000 lb.
Iron rivets in double shear	70,000 lb.
Steel rivets in single shear	42,000 lb.
Steel rivets in double shear	78,000 lb.

Factors of Safety

5. The lowest factors of safety used for steam boilers the shells or drums of which are directly exposed to the products of combustion, and the longitudinal joints of which are of lap-riveted construction, shall be as follows:

(*a*) Five (5) boilers not over ten years old.

(*b*) Five and five-tenths (5.5) for boilers over ten and not over fifteen years old.

(*c*) Five and seventy-five hundredths (5.75) for boilers over fifteen and not over twenty years old.

(*d*) Six (6) for boilers over twenty years old.

(*e*) Five (5) on steam boilers the longitudinal joints of which are of lap-riveted construction, and the shells or drums of which are not directly exposed to the products of combustion.

(*f*) Four and five-tenths (4.5) on steam boilers the longitudinal joints of which are of butt and strap construction.

Section 2

Section 2 sets forth the standard form of certificate of annual inspection.

Section 3 — Fusible Plugs

1. Fusible plugs, as required by section 20, chapter 465, Acts of 1907, shall be filled with pure tin.

2. The least diameter of fusible metal shall not be less than one-half ($\frac{1}{2}$) inch, except for working pressures of over one hundred and seventy-five (175) pounds gage or when it is necessary to place a fusible plug in a tube; in which cases the least diameter of fusible metal shall not be less than three-eighths ($\frac{3}{8}$) inch.

3. The location of fusible plugs shall be as follows:

(*a*) In Horizontal Return-tubular Boilers — In the back head, not less than two (2) inches above the upper row of tubes, and projecting through the sheet not less than one (1) inch.

(*b*) In Horizontal Flue Boilers — In the back head, on a line with the highest part of the boiler exposed to the products of combustion, and projecting through the sheet not less than one (1) inch.

(*c*) In Locomotive Type or Star Water-tube Boilers — In the highest part of the crown sheet, and projecting through the sheet not less than one (1) inch.

(*d*) In Vertical Fire-tube Boilers — In an outside tube, placed not less than one-third ($\frac{1}{3}$) the length of the tube above the lower tube-sheet.

(*e*) In Vertical Submerged-tube Boilers — In the upper tube-sheet.

(*f*) In Water-tube Boilers, Horizontal Drums, Babcock & Wilcox Type — In the upper drum, not less than six (6) inches above the bottom of the drum and over the first pass of the products of combustion, projecting through the sheet not less than one (1) inch.

(*g*) In Stirling Boilers, Standard Type — In the front side of the middle drum, not less than six (6) inches above the bottom of the drum, and projecting through the sheet not less than one (1) inch.

(*h*) In Stirling Boilers, Superheated Type — In the front drum, not less than six (6) inches above the bottom of the drum, and exposed to the products of combustion, projecting through the sheet not less than one (1) inch.

(*i*) In Water-tube Boilers, Heine Type — In the front course of the drum, not less than six (6) inches from the bottom of the drum, and projecting through the sheet not less than one (1) inch.

(*j*) In Robb-Mumford Boilers, Standard Type — In the bottom of the steam and water drum, twenty-four (24) inches from the center of the rear neck, and projecting through the sheet not less than one (1) inch.

(*k*) In Water-tube Boilers, Almy Type — In a tube directly exposed to the products of combustion.

(*l*) In Vertical Boilers, Climax or Hazelton Type — In a tube or center drum not less than one-half ($\frac{1}{2}$) the hight of the shell, measuring from the lowest circumferential seam.

(*m*) In Cahall Vertical Water-tube Boilers — In the inner sheet of the top drum, not less than six (6) inches above the upper tube-sheet.

(*n*) In Scotch Marine Type Boilers — In combustion-chamber top, and projecting through the sheet not less than one (1) inch.

(*o*) In Dry-back Scotch Type Boilers — In rear head, not less than two (2) inches above the top row of tubes, and projecting through the sheet not less than one (1) inch.

(*p*) In Economic Type Boilers — In the rear head, above the upper row of tubes.

(*q*) In Cast-iron Sectional Heating Boilers — In a section over and in direct contact with the products of combustion in the primary combustion chamber.

(*r*) For other types and new designs, fusible plugs shall be placed at the lowest permissible water level in the direct path of the products of combustion, as near the primary combustion chamber as possible.

XVIII

MECHANICAL TUBE CLEANERS

THE Hartford Inspection and Insurance Company, in a recent issue of *The Locomotive*, sounds a note of alarm anent the damage which may be inflicted upon a boiler by the improper use of mechanically operated tube cleaners. Coming, as it does, from so high an authority, this warning has produced unnecessary alarm among the present or prospective users of such devices, and the statement that the dangers pointed out are incident to them when improperly handled, and that "many of them give very good results when used judiciously and intelligently," is lost sight of in the light of the stated fact that injury has been produced by their use.

The first instance pointed out is one in which by the use of a cleaner removing external scale by rapidly rapping the internal surfaces of the tubes the latter were stretched to an elliptical section to such an extent that several of them collapsed when subjected to a pressure of ninety pounds. With any of the cleaners as now built by experienced and reputable makers such a result could be produced only by the grossest misuse of the tool and the most flagrant neglect of the directions which are furnished with it. Tests made

by Professor Kavanaugh, of the University of Minnesota, with a $3\frac{1}{2}$-inch cleaner prove the energy of the blow when operating under a pressure of 90 pounds to be .106 of a foot-pound and the number of blows per minute 4,560. Only slight local distortion was produced by allowing the hammer to operate continuously in one spot.

It appears, therefore, that the distortion of the tubes in the case mentioned must have been due to a very unskilful use of a very badly designed cleaner rather than to the fact that the tubes were thinned by use but still serviceable. The same remarks will apply to the cases of splitting mentioned.

Another effect is the lengthening of the tubes due to the peening action, causing them either to sag or to project through the head. This action might follow an unduly protracted application of even a good cleaner, but should not be caused by such application as is necessary to remove ordinary scale. Such elongation is liable to crack the cast-iron headers of water-tube boilers, and the makers of at least one of the cleaners of this type discourage for this reason its use in boilers with headers of that material. Such headers are, however, dangerous in themselves and their use is rapidly being discontinued. This peening effect should be present in the mind of the operator of the cleaner, and he should regulate the intensity and time of application of the blows so as to avoid it, and watch carefully for it at the tube sheets.

The unequal expansion caused by discharging the exhaust from steam-operated cleaners through the tubes

is also considered, and the use of compressed air for running the cleaner, when available, advised. The makers of the cleaners are alive to this condition. recommend the use of air in preference to steam, and recommend also that the boiler be cleaned while hot.

The conclusions arrived at in *The Locomotive* article are as follows:

(1) That when power-tube cleaners are used they should be kept in motion so that they cannot strike a succession of blows against any one part of the tube; (2) they should be operated by a pressure not exceeding 20 pounds, or, at the most, 30 pounds per square inch; (3) steam should not be permitted to blow through the tubes of a cold boiler for a sufficient time to sensibly heat the tubes; (4) compressed air should be used to operate tube cleaners unless the motive power is entirely external to the tube; (5) in any case, the boiler should be carefully watched during and after the application of a power cleaner, especially around the ends of the tubes and on the headers, and at the first sign of distress of any kind the use of the cleaner should be promptly discontinued; (6) lastly, a power cleaner should never be put in charge of any attendant save one upon whose judgment and skill the owner of the boiler can implicitly rely.

These conclusions commend themselves even to the makers of the devices in question, with the exception that they claim that a pressure of from 40 to 90 pounds is better than the lower pressure recommended as giving more rapid vibrations and of less amplitude, and deny that the heating due to exhaust steam will injure a

sound tube. Signs of distress may be evidences of weakness revealed by the cleaner, and point to reforms or repairs rather than the discontinuance of the use of the cleaner. The strictures apply principally to cleaners operating by hammer action and discharging steam through the tube, but do not amount to a condemnation of the type, the successful present use of over 5,000 machines for a single maker evidencing that injury from its use is exceptional and avoidable rather than general and inherent.

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PIPES AND PIPING

PREFACE

THIS book gives general rules for the design of high and low pressure steam piping in power plants. It gives the reader ideas as to forces to be met and the amount of resistance that can be expected from the pipe and fittings properly placed. An idea of good steam-pipe design for all size plants is given together with many useful suggestions as to the installation and operation of the same pipe installation.

The material here collected is selected from the columns of Power, where it has appeared from time to time, and the author is greatly indebted to the contributors and editors for valuable material and suggestions.

HUBERT E. COLLINS.

NEW YORK, September, 1908.

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GENERAL DESIGN

I

THE successful operation of steam mains for high-pressure steam work requires, (1) good design; (2) good material; (3) skilful steam fitters. Of the first requirement there can be many modifications to suit conditions, but a few fundamental facts here stated will be of use for general application.

Up to the year 1900 the almost universal design for a piping system for power plants was to carry a pipe for each boiler into a large steam header, whose cross-section was equal to the sum of all the areas of the feeding pipes. If the designing engineer wanted to be very grand, he made it larger without knowing just why. From this header pipes were carried to the various engines, of sizes called for by the engine builder. It was quite common to place steam separators in each line leading to an engine, but fashions changed in steam-piping as well as in clothes, and with the advent of high pressure and superheated steam sizes of pipes have been very much reduced. In the four large power-houses now built in New York City, with an ultimate capacity of from 60,000 to 100,000 horse-power each, the largest steam mains are not over 20 inches in diameter, and these are used

more as equalizing pipes than storage reservoirs. Some of our best plants have pipes which run from the header to the engine two sizes smaller than that called for by the engine builders. These pipes, before reaching the engine, are carried into a wrought-iron or steel receiver, which acts also as a separator. This receiver has a cubical capacity three times that of the high-pressure cylinder, and is placed as near as possible to the cylinder. The pipe from the receiver to the cylinder is of the full size called for by the engine builder. The object of this arrangement is, first, to have a full supply of steam close to the throttle; second, to provide a cushion near the engine on which the blow caused by the cut-off in the steam-chest may be spent, thereby preventing vibrations from being transmitted through the piping system, and, third, to produce a steady and rapid flow of steam in one direction only, by having a small pipe leading into the receiver. This steam flows rapidly enough to make good the loss caused during the first quarter of the stroke. Plants fitted up in this way are successfully running where the drop in steam pressure is not greater than four pounds, although the engines are 500 feet away from the boilers. This is probably the most radical departure in high-pressure work up to the present time.

In estimating the size of pipe desired for a given size of cylinder of a reciprocating engine, a prominent designer uses the following formula:

$$\frac{\text{Area of cylinder}}{\text{Velocity of steam in pipe} \div \text{piston speed}} = \text{area of steam-pipe.}$$

Example: simple engine 16″ cylinder, 30″ stroke, 150 r.p.m., velocity of steam 8000 feet per min. What size of pipe is required?

Area of 16″ cylinder	= 201″ sq.
$30'' \times 2 \times 150 \div 12$	= 750 feet piston speed.
$8000 \div 750$	= 10.6 ratio of flow.
$201 \div 10.6$	= 18.9 sq. ins. area of steam-pipe.
18.9 sq. in.	= 4.35″ dia. of steam-pipe.
Nearest dia.	= $4\frac{1}{2}''$ pipe.

In figuring sizes of pipe for steam-engines the following holds good in practice.

Diameter of steam-pipe for a simple engine = .35 of cyl. dia.

Diameter of exhaust pipe for a simple engine = .40 of cyl. dia.

Diameter of steam-pipe for a compound engine = .40 of L P cyl. dia.

Diameter of exhaust pipe for a compound engine = .40 of L P cyl. dia.

These proportions are ample and are those used in engine practice.

EXPANSION AND CONTRACTION IN STEAM PIPES*

In laying out a system of steam-piping for a power plant perfect freedom for expansion and contraction should be allowed, to prevent undue strains on any member of the system or at the joints, causing them to leak. The old types of slip-expansion joints having proved a constant source of trouble and expense requiring frequent repacking and adjusting, are seldom, if ever, used on a good job of piping. If absolutely necessary, however, to use this type of expansion joint, the piping should be so anchored as to prevent the joint from pulling apart.

With the advent of higher steam pressures and correspondingly higher temperatures and velocities, more attention has been given to the proper designing of piping systems and constructive details than in the past. Steel pipe bends of long radius are used wherever practical in place of the cast elbows of short radius. They take up the expansion stresses, making the system more flexible throughout; reduce vibrations and friction, and deliver the steam to the engine with a lower drop in pressure.

Pipe bends curved to a radius of less than five diameters of the pipe are undesirable as expansion

*Contributed to Power by William F. Fischer.

bends, because, being so stiff, they fail to take up the expansion in the line, and a good deal of strain is thrown on the fittings and joints. The radius should be at least five or six pipe diameters and if possible even greater than six diameters, say ten or twelve, in order to give sufficient elasticity.

Anchoring

The method of anchoring the steam line and the position of the anchor are important details. The line may be so anchored as to throw severe strains on the joints, working the gaskets and bolts loose and causing leakage; or a broken fitting or cracked flange may be the result, making it necessary to cut out one or more units while making repairs. Many leaky joints can be traced to this cause.

No special rules can be given for designing and anchoring for expansion. The designer must depend on his own good judgment in placing the bends, anchors, etc., where they will do the most good. However, a knowledge of the expansion of pipes when heated is essential.

Expansion

In Fig. 1 there is shown a 10-inch extra heavy steam-pipe, and two clamps *A A* anchored rigidly to the walls. Say, for example, the pipe was placed in the position shown, at a temperature of 60 degrees Fahrenheit, anchored rigidly between one clamp and allowed to move freely through the other when expanding; supposing that while in this position, steam at 250 pounds pressure (above vacuum) was turned into it

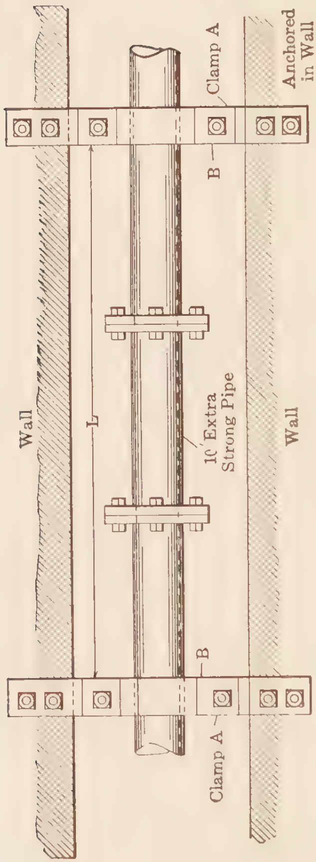

Wall
Clamp A
Anchored
in Wall
B
L
10″ Extra
Strong Pipe
Wall
B
Clamp A

FIG. 1

superheated 100 degrees, the pipe would heat and cause it to expand and slide through the clamp at the free end.

The elongation of the pipe, due to expansion due to the difference of the above temperatures, can be calculated by the following formula:

$$e = LCT \tag{1}$$

where

e = Amount of expansion or contraction, in inches, due to heating or cooling of pipe;

L = Length of pipe, in inches, at its original temperature before heating or cooling;

T = Difference in degrees Fahrenheit of temperature between the original temperature of the pipe and its temperature after being heated or cooled;

C = Coefficient of linear expansion.

We have assumed the original temperature of the pipe to be 60 degrees Fahrenheit. In the steam tables we find the temperature of steam at 250 pounds above vacuum to be 401 degrees Fahrenheit. Adding 100 degrees of superheat we obtain

$$401 + 100 = 501$$

degrees Fahrenheit, which equals the temperature of the steam flowing through the pipe. Assuming the pipe to be heated to the same temperature as the steam, then

$$T = 501 - 60 = 441$$

degrees Fahrenheit.

Assume the length L, or distance between the clamps to be 60 feet, then

$$L = 60 \times 12 = 720$$

inches.

The coefficient of expansion C can be found in almost any engineers' handbook for different materials.

For wrought iron it is usually taken as 0.00000686; for untempered steel and cast iron as 0.000006; for brass as 0.00001 and for copper as 0.000009.

By substituting the above values for L and T, and taking C for steel pipe as 0.000006 we obtain in formula (1)

$$e = L\,C\,T,$$

or

$$e = 720 \times 0.000006 \times 441 = 1.9,$$

or $1\frac{7}{8}$ inches.

That is, the original length of the pipe has been increased from 60 feet to 60 feet $1\frac{7}{8}$ inches, due to the added temperature of 441 degrees, and the movement of the pipe through the loose clamp is $1\frac{7}{8}$ inches.

Force of Expansion

If we assume the pipe to be clamped between both clamps while at its original temperature, 60 degrees Fahrenheit, in expanding when heat is added it would exert a thrust against the clamps as shown by the leaders B B, and either the clamps would spring, or the pipe would buckle or bend sidewise, owing to its great length as compared with the diameter.

We can calculate fairly correctly the magnitude of this thrust by the following three formulas: Formula

(1), as used for expansion and contraction, and formulas (2) and (3), which are used in finding the elongation of a bar of metal due to a given external force. In formula (1)

$$e = L C T,$$

the rotation being the same as previously used.

In the formula

$$e_1 = \frac{PL}{AE} \tag{2}$$

and in the formula

$$P = \frac{e_1 AE}{L} \tag{3}$$

e = Total elongation of body in inches;
A = Area of metal of a cross-section of body in square inches;
P = Total stress on body in pounds;
L = Total original length of body in inches;
E = Coefficient of elasticity of metal composing body.
E for wrought iron = 25,000,000;
E for steel = 30,000,000;
E for cast iron = 15,000,000.

In formulas (2) and (3) e is equivalent to e in formula (1) with the exception that the elongation e in formula (1) is due to an internal force due to heating the pipe, and the elongation e, formulas (2) and (3), is caused by an external force, such as a weight, or pull.

If we substitute e in formula (1) for e in formula (3) we get

$$P = \frac{eAE}{L},$$

and by substituting for e its equivalent $L\,C\,T$ we get

$$P = \frac{LCTAE}{L},$$

and by cancelling L we have

$$P = C\,T\,A\,E, \tag{4}$$

where

P = Magnitude of thrust in pounds exerted by the pipe when expanding, or the pull when contracting;

C = Coefficient of linear expansion;

E = Coefficient of elasticity;

A = Area of metal in cross-section of pipe in square inches;

T = Difference in degrees Fahrenheit of temperature between the original temperature of pipe and its temperature after being heated or cooled.

Thus, formula (4) should give us the magnitude of the thrust exerted by the pipe as it expands. This formula should be used only where approximate close results are required, and within given temperatures only, as a body when heated beyond a certain temperature loses a large percentage of its strength.

This formula may be used for wrought-iron and steel steam-pipes heated up to 600 degrees, and as we seldom need go higher than this, we will not consider it above this temperature.

As an example, let us find the thrust P exerted on

the clamps, in Fig. 1, due to the expansion of the pipe, using the same dimensions and temperatures as before. Then

C = 0.000006;
E = 30,000,000;
T = 441 degrees;
A = Area of metal of a cross-section of pipe.

This is found in the National Tube Company's handbook to be 16 square inches for a 10-inch extra-heavy pipe.

By substituting the above in formula (4) we have

$$P = C\,T\,A\,E,$$

or

$P = 0.000006 \times 441 \times 16 \times 30{,}000{,}000 = 1{,}270{,}080$ pounds.

This gives some idea of the strains thrown on the fittings and joints where no provision is made to take up the expansion and contraction in steam lines. If anchored improperly either the anchors would give, or the pipe would spring sidewise, straining the joints sufficiently in many cases to cause excessive leakage, or to crack the flanges.

Contraction

If, in Fig. 1, the pipe were heated and allowed to expand freely through the clamps, and then clamped tightly, and allowed to cool off again, it would shrink or contract, subjecting the joints to a tensile strain of 1,270,080 pounds. Assuming the anchors to be of

sufficient strength, and rigid enough to resist bending, the tensile strain on the material in the pipe would be

$$\frac{P}{A} = \frac{1,270,080}{16} = 80,000$$

pounds per square inch; sufficient to cause rupture.

Allowing for Expansion and Contraction

It is considered good practice in figuring for expansion to allow only one-half the calculated amount when cutting the pipe to length. For example, if in a run of pipe 100 feet between connections, or points where steam lines are taken off from the header, the expansion is calculated to be 3 inches; allow $1\frac{1}{2}$ inches when cutting the pipe to length, or in other words, the total length of pipe should be $100 - 1\frac{1}{2} = 99$ feet $10\frac{1}{2}$ inches.

Then the steam fitter takes up the other $1\frac{1}{2}$ inches when erecting the line, with the result that when steam is turned on the expansion removes the tension or strain put on the pipe when cold, and the fittings and joints are strained only one-half as much as if none, or all of the expansion were allowed for. This rule is used by most large concerns, and has proven to be satisfactory to operating conditions in general.

Taking Care of the Expansion

In Fig. 2 a good example of a connection between the boilers and main steam header is shown. The bend is designed to take up the expansion and contraction in both the main steam header and the con-

nection from the boilers to the header. With this arrangement the expansion strains on the fittings and joints are greatly reduced. The arrows show the movement of the pipe due to expansion and contraction.

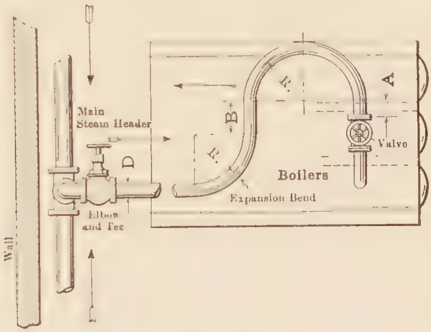


FIG. 2

This arrangement is particularly desirable where the boilers are placed in rows along the division wall, and the main steam header runs parallel with the wall, as shown.

As mentioned before, to have sufficient elasticity, the radius *R* should not be less than five diameters of *D*. The length of the straight portion of pipe at *A* on the end should be at least one diameter of *D;* with sizes from 5 inches and upward one and one-half diameters are preferred, and not less than 4 inches for smaller sizes.

The length of the straight pipe at *B* between the arcs should be from 6 to 12 inches, or greater if preferred. This insures better bends and prevents kinks when bending, which usually occur when no straight

pipe is allowed, particularly on large pipe, also making the bend more flexible.

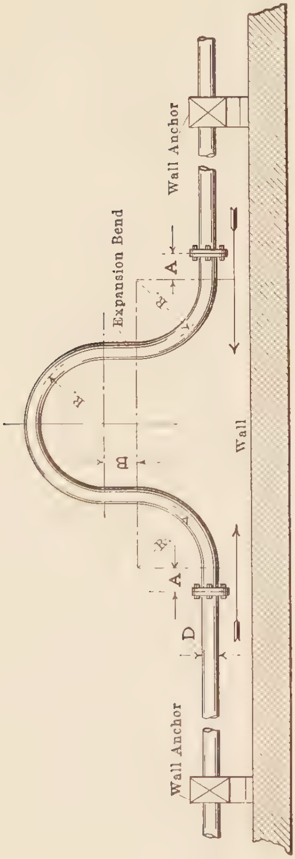
The expansion bend shown in Fig. 3 is intended for use in a long run of piping where it is desirable to anchor the line at two points to take the strain off the fittings and joints outside the anchors, not shown. When heated, the pipe expands in the direction of the arrows, tending to close the bend. To insure sufficient elasticity the dimension *B* should be made as large as possible. The dimensions *A* and *R* can be the same as in Fig. 2.

For sizes above 6 inches it may be impossible to make this bend in one piece, owing to the long length of pipe required. Commercial pipe averages from 16 to 20 feet in length, although lengths up to 24 feet are kept in stock by some dealers for use in making up special bends.

For bends 8 inches in diameter and larger, the arrangement shown in Fig. 4 may be used for the same purpose. With such an arrangement, however, a bending strain is put on the joints, as shown in Fig. 5, which is greatly enlarged to show the straining action on the joint at *E*, as the pipe expands.

The cast elbows should be extra heavy, with thick flanges. The flanges on the bend should be thick also, as a thin flange is easily ruptured or strained sufficient to cause leakage at the joints.

Where there is sufficient room, a better arrangement would be to substitute two 90 degree or square bends of steel pipe in place of the cast elbows, making the system more flexible.



For Long Runs
FIG. 3

The method of anchoring a steam header of moderate length, say 50 to 125 feet, is shown in Fig. 6. The line is anchored rigidly at the center and allowed to expand both ways, as indicated by the arrows *A*. The joints nearest the anchor are subjected to the least strain, the greatest strain falling on the joints near the ends of the header.

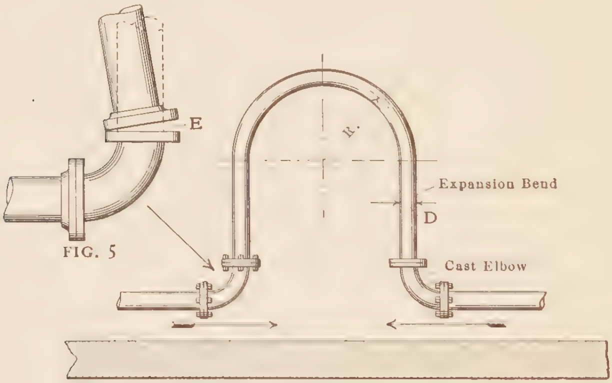


FIG. 5

FIG. 4

Proper provision should be made in connecting up to the boilers and engines to take care of this expansion, and before doing so it is a good plan to figure up the different runs of piping and work out the amount of expansion in each case.

This will give a better idea of the conditions, and proper provision can be made to handle each case accordingly.

The expansion of cast iron in fittings is about the same as untempered steel, but under the strain and

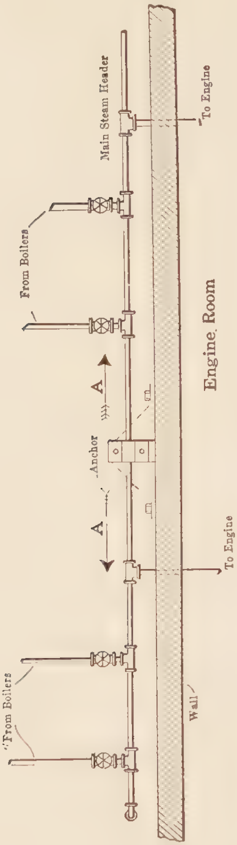

Main Steam Header
To Engine
From Boilers
A
Anchor
A
To Engine
Engine Room
Wall
From Boilers

FIG. 6

stress of superheated steam (heat and pressure combined) a fitting 22 inches in length from face to face of flanges has been known to get a permanent elongation of $\frac{5}{16}''$ in four years' use. Therefore in the design of steam-pipe lines, the question of superheat must be considered, not only for expansion alone, but for the permanent elongation and the loss of tensile strength of the material.

II

ESTIMATING STOCK FOR CURVED PIPES*

The length of pipe necessary for curved work can be estimated in two ways, and the method to use, or whether a combination of both methods is necessary,

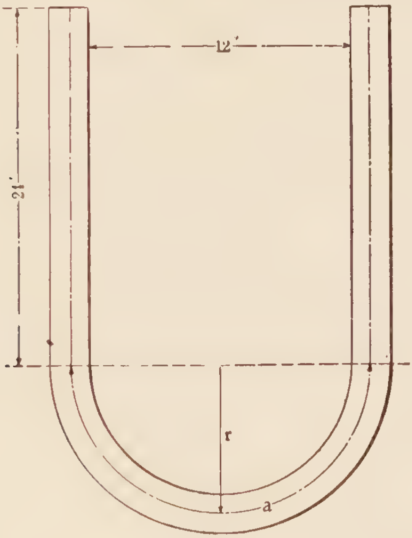


FIG. 7

will depend upon the form of the curves. Thus, take the simple U-bend shown in Fig. 7, where all the

* Contributed to Power by F. Webster.

dimensions of the finished work are known. The length of stock necessary will be approximately that of the length of the dotted line along the center of the pipe. This line is made up of two straight portions and a curve which is one-half the circumference of a circle. To get the circumference of any circle, multiply the diameter by the constant 3.1416, and for the length of a semicircle, as *a*, Fig. 7, multiply the radius *r*, which is one-half the diameter, by 3.1416. Wrought-iron pipes are designated by the length of the diameter of the hole. When making bent-pipe work, it is often necessary to know the outside diameter of the pipe. The dimensions of pipe are given in tables, such as the accompanying, and it is most convenient to use the tables when making estimates. There are, however, slight variations in the dimensions of pipes, so that those given in the table do not always strictly apply, but at the same time the dimensions given in the table are close enough for any work that the blacksmith may be called upon to make.

The solution of the example shown in Fig. 8, where 1-inch pipe is used, will illustrate the method. From the table the outside diameter of the pipe is 1.31 inches. The radius *r* of the curve equals

$$\frac{15}{2} + \frac{1.31}{2},$$

$$\frac{16.31}{2}$$

The length of the central or neutral lines of the curve equals

$$\frac{16.31}{2} \times 3.1416 = 25\tfrac{1}{2} \text{ inches.}$$

The total length of the stock equals $50\frac{1}{2}$ inches.

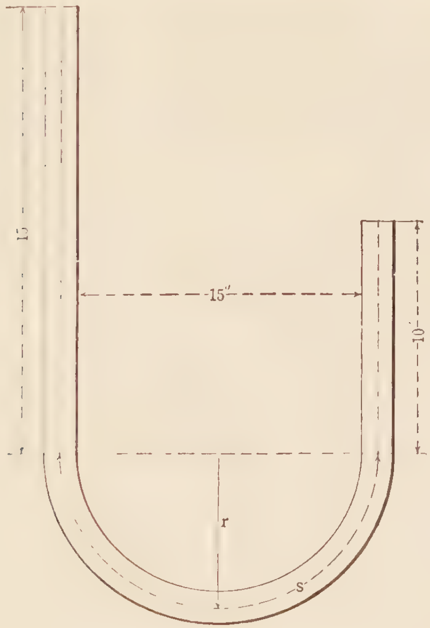


FIG. 8

For making a right-angle bend, as shown in Fig. 9, the length of the arc depends upon the length of the radius r used. Suppose $r = 6$ inches, and 1-inch pipe is used, the solution is then as follows:

The radius R of the center line of the pipe equals

$$6 + \frac{1.31}{2} = \frac{13.31}{2},$$

and the length of the arc equals

$$\frac{13.31}{2} \times \frac{3.1416}{2} = 10\tfrac{1}{2} \text{ inches.}$$

The length of the straight piece *a* equals

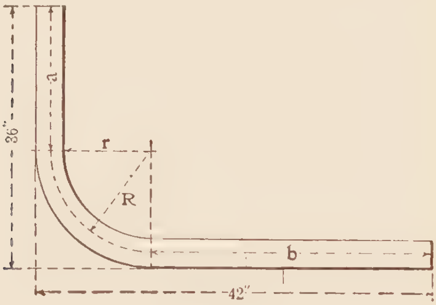


FIG. 9

$$36 - (6 + 1.31) = 28.7 \text{ inches,}$$

and the length of *b* equals

$$42 - (6 + 1.31) = 34.7 \text{ inches.}$$

The total length equals

$$10.5 + 28.7 + 34.7 = 74 \text{ inches,}$$

approximately.

To estimate the length of pipe required for the work shown in Fig. 10, make an accurate drawing of the work to some definite scale, and measure the length of

the dotted center line, using a tape-line or cord, or a measuring wheel, and multiply the length by the scale.

All bends should be made up with the seam on the inside radius.

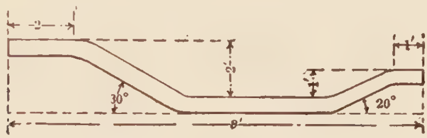


FIG. 10

The length of pipe necessary to make up a bend as shown in Fig. 3, Chapter I will be

$$L = 2 \times R \times 3.1416 + 2 \times A + 2 \times B = \text{length in feet.}$$

Then for a 6″ bend taking A and B equal to one diameter of D, and R equal to 6 diameters, an advisable radius.

$$L = 2 \times 3 \times 3.1416 + 2 \times 0.5 = 20.85 \text{ feet or } 20' \; 10\tfrac{3}{16}''.$$

III

VIBRATION IN STEAM-PIPES*

VIBRATION in steam-pipes is usually due to faulty design. Steam-pipes are generally so designed that the flow is about a mile a minute, and in steam-turbine work about a mile and a half a minute. Where in the piping there are a number of sharp turns through short-radius elbows, steam traveling at this high velocity is very likely to set up vibration in the line due to the sudden change in the direction of the flow. Even if the line is anchored it does not always cure the vibration, as the cause remains and in time the anchors may become loosened sufficiently to allow the pipe to vibrate as badly as before. Excessive vibration causes the joints to leak by working the bolts loose and taking the tightening pressure off of the gasket. In screwed work, where the pipe is screwed through the flange, or inso a screwed fitting, it sometimes causes leakage through the threads.

Figure 11 shows a method very often employed in connecting up an engine at the end of a steam line. Here *A* and *B* are short-turn elbows. From the elbow *B* the line drops to the high-pressure cylinder through the engine throttle-valve. Excessive vibrations oc-

* Contributed to Power by William F. Fischer.

curred in a line exactly similar to this, and were finally obviated by arranging the piping as shown in Fig. 12. The engine was of the slide-valve type; a separator in the main line took care of several engines.

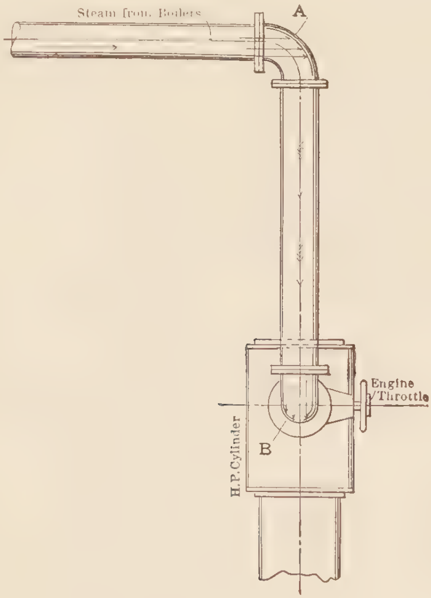


FIG. 11

What Caused the Vibration?

The following simple explanation will probably make this clear. Suppose, for example, that steam is forced out of the end of a straight, unobstructed pipe, as shown in Fig. 13. As long as the pressure in the line

remains constant the steam flows from the nozzle at a constant velocity, causing very little, if any, vibration in the line. Suppose the pipe to be equipped at the end with a quick-opening and closing valve, capable of being opened and closed without jar or shock to the piping. While the steam is flowing the valve is sud-

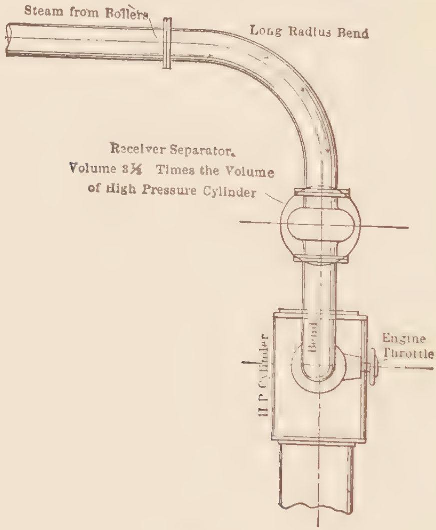


FIG. 12

denly closed. What happens? The rapidly moving steam, driven by the pressure behind it, strikes the valve-seat a quick blow (due to its momentum) tending to pull the line with it in the direction of the flow. The pipe then tends to spring back again to its original position. When the steam in front strikes the valve-

seat, the steam coming on behind tends to pile up on the steam in front until all the steam in the line, back to the source of supply, is brought to rest. This occurs very quickly.

Suppose the valve to be opened and closed, say 180 times a minute, as in the engine in the illustration. Steam would strike the valve-seat each time the valve closed, tending to pull the line with it, and each time

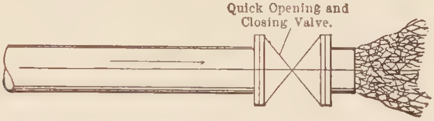


FIG. 13

there would be a reaction. This would naturally tend to impart motion to the pipe if it were free to move even slightly. This rapid motion is what constitutes the vibration. The opening and closing of the slide valve in the steam-chest gives practically this same result, with the exception that the steam is not traveling as fast as before, due to the pressure required to force the piston outward, therefore, the force of the blow would not be as great. Still, it is sufficient to cause vibration in many cases.

Setting the engine valves improperly has a tendency to cause vibration also which is transmitted through the piping.

The short turn elbows *A* and *B*, Fig. 11, help along the vibration. The steam, in stopping and starting up to full speed again, when the engine valve is opened and closed, strikes the elbow *A* a glancing blow on the

side, as indicated by the arrows, before being deflected at right angles, and strikes the elbow *B* again before being deflected downward. This has a tendency to set up a slight but rapid vibration.

By substituting long-radius bends of steel pipe in place of the two elbows *A* and *B*, and placing a separator of large volume in the line, as near the engine cylinder as possible, the vibrations are eliminated, at least to such an extent as not to be noticeable in the line, or cause annoyance.

Receiver-Separators Prevent Vibration

In well-designed piping systems, receiver-separators of large volume are nearly always used, placed close to the engine cylinder. Where such separators have a capacity of three times the volume of the high-pressure cylinder, or greater, the piping may be reduced 10 to 15 per cent. from the sizes called for by the engine manufacturers, this reduction being made in the piping at the inlet side of the separator. The action of the steam going through the separator is somewhat as follows:

When the valve opens at each stroke of the engine piston, the engine obtains the necessary volume of steam from the separator to force the piston to the end of the stroke. This requires but a fraction of a second in ordinary cases. In case of the engine in Figs. 11 and 12, making 90 revolutions, or 180 strokes per minute, it requires $60 \div 180 = \frac{1}{3}$ of a second, neglecting the point of cut-off. In this short space of time enough steam is drawn from the separator to reduce pressure.

The boiler pressure is forcing new steam to the separator, through the inlet pipe, at a high velocity, and when the valve closes again, the steam rushes into the separator, crowding the other steam together, and restoring the volume and pressure required at the next stroke of the piston. This goes on continuously while the engine is running. The steam from the inlet pipe to the separator flows at an almost constant velocity, cushioning itself against the steam already in the separator, thus causing a steady and rapid flow of steam to the engine and preventing vibration.

Referring to Fig. 12 again, it is easy to understand why this arrangement prevents vibration. It removes the cause. The separator acts as a reservoir in which the steam is cushioned after each stroke of the engine; or in other words, when the valve closes, the oncoming steam tends to pile up in the separator, surging in and out as the valve opens and closes. This reduces the shock by taking the reaction caused by the quick cut-off in the steam chest. It removes most of the moisture, and if a slug of water is driven down, it is caught up by the separator and thrown to the bottom of the well, thus preventing it from getting into the engine cylinder. The two long-radius bends turn the steam gradually from its straight course bringing it down to the engine cylinder without jar or shock.

The bends should be large radius, not under five times the pipe diameter; 8 to 10 diameters and even greater are preferable. Long-radius bends also reduce the friction in the line, giving a higher velocity and pressure at the engine cylinder.

IV

HIGH-PRESSURE STEAM-PIPE FLANGES *

THE need of a good high-pressure flange joint becomes every day more urgent, as there is the greatest difference between the character of steam to-day and that used several years ago, making it practically impossible to use "standard" flanges, suitable at one time, for a pressure of 120 and 150 pounds per square inch. In an up-to-date plant it is not remarkable to find pressures up to 225 pounds, and sometimes even higher. It is sufficient to point to the French six-cylinder quadruple-expansion engine at the World's Fair at St. Louis, running at a pressure of some 300 pounds per square inch, while the steam was superheated to a temperature of 750 degrees Fahrenheit. Of course, this extremely high pressure and degree of superheat affects the entire pipe system, and not alone the joint; but it is my purpose here to discuss the latter only.

Figure 14 shows the Allen patent loose flange which can be used on pipe from 1 inch diameter up. The accompanying table of the "Verein Deutscher Ingenieure" is shown in Fig. 15. Here one will find pipe flanges of from 1 to 16 inches in diameter, and the

* Contributed to Power by Franz Koester and Luther D. Lovekin.

larger sizes are merely additions in the amount of material. This table gives the flange standards of the German Society of Engineers, and as they were therefore dimensioned on the metric system, the author has not only converted their figures, but laid out each individual flange so as to get even fractions of an inch. The number and sizes of bolts have not been changed, and it will be noticed that there are flanges

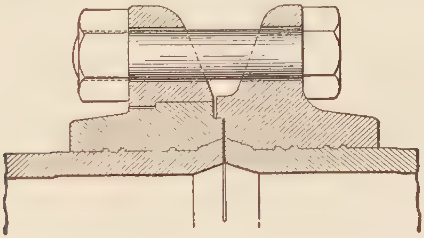
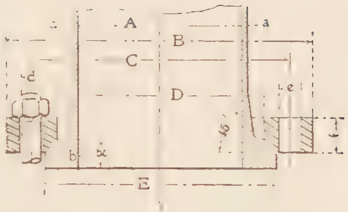
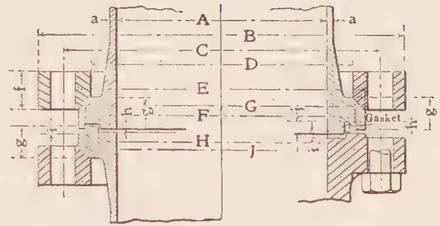


FIG. 14

provided with 6, 10, 14 and 18 bolts, in each case a number not divisible by 4, the latter being much favored in power-plant design. This may be due to the fact that with loose flanges it is not so necessary to straddle the centers. Of course, this does not apply so much to joints on valves, cylinders or any other cast-iron sections.

The many advantages of loose-flange joints, such as a sure fit of bolt-holes and the possibility of shifting the pipes smaller distances than the arc between the consecutive holes; the easy assembling of the joints, and the easy inspection, as the flanges in many designs do not come so closely together, induced the engineers





Size	A	1	1½	2	2½	3	3½	4	4½	5	6	7	8	9	10	11	12	13	14	15	16
Thickness of shell	a	$\frac{3}{32}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	[illegible]	$\frac{7}{32}$	$\frac{3}{16}$	[illegible]	$\frac{7}{32}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{9}{32}$	$\frac{9}{32}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{11}{32}$	$\frac{11}{32}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$
Thickness of shoulder	b											$\frac{1}{2}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$
Outside diameter of loose flange	B	$4\frac{7}{8}$	$5\frac{7}{8}$	$6\frac{1}{4}$	7	$7\frac{3}{4}$	$8\frac{3}{4}$	$9\frac{1}{2}$	10	$10\frac{5}{8}$	$11\frac{3}{4}$	13	$14\frac{1}{4}$	$15\frac{5}{8}$	$16\frac{3}{4}$	18	$19\frac{1}{8}$	$20\frac{1}{2}$	22	$23\frac{1}{8}$	$24\frac{1}{4}$
Diameter of bolt circle	C	$3\frac{3}{4}$	$4\frac{3}{8}$	5	$5\frac{1}{2}$	$6\frac{1}{4}$	7	$7\frac{1}{2}$	8	$8\frac{5}{8}$	$9\frac{3}{4}$	11	$12\frac{1}{4}$	$13\frac{1}{2}$	$14\frac{5}{8}$	$15\frac{3}{4}$	11	$18\frac{3}{8}$	$19\frac{5}{8}$	$20\frac{3}{4}$	$21\frac{3}{4}$
No. of bolts		4	6	6	6	6	6	6	8	8	8	10	12	12	12	14	16	16	16	18	20
Size of bolts	d	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	1	1	1	1	$1\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{1}{8}$
Diameter of bolt-holes	e	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	1	1	1	$1\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$
Inside diameter of loose flange	D	$2\frac{3}{8}$	3	$3\frac{3}{8}$	$3\frac{3}{4}$	$4\frac{1}{2}$	$4\frac{7}{8}$	$5\frac{1}{2}$	$6\frac{1}{2}$	$6\frac{1}{2}$	$7\frac{3}{4}$	$8\frac{7}{8}$	10	$11\frac{1}{4}$	$12\frac{3}{4}$	$13\frac{3}{8}$	$14\frac{1}{2}$	16	$17\frac{1}{8}$	$18\frac{1}{4}$	$19\frac{1}{4}$
Thickness of loose flange	f	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{7}{8}$	1	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{5}{8}$	$1\frac{3}{4}$	$1\frac{7}{8}$	2	$2\frac{1}{8}$	$2\frac{1}{4}$	$2\frac{3}{8}$	$2\frac{1}{2}$

Diameter of pipe flange	E	3	3 3/8	4	4 1/2	5 1/4	6	6 1/2	7	7 1/2	8 5/8	9 7/8	11 1/8	12 1/4	13 3/8	14 1/2	15 3/4	17	18 1/4	19 3/8	21 3/4
Thickness of pipe flange	g	5/8	3/4	3/4	7/8	7/8	1	1	1 1/8	1 1/4	1 1/4	1 1/4	1 1/4	1 3/8	1 1/2	1 1/2	1 1/2	1 5/8	1 5/8	1 5/8	1 3/4
Inside diameter of groove....	F	1 7/8	2 3/8	2 7/8	3 3/8	4	4 1/2	5	5 1/2	6	7 1/4	8 1/4	9 1/2	10 1/2	11 3/4	12 3/4	13 3/4	15	16 1/4	17 1/4	18 1/4
Outside diameter of groove....	G	2 3/8	3 1/8	3 5/8	4 1/8	4 3/4	5 3/8	5 7/8	6 3/8	6 7/8	8 1/8	9 1/4	10 1/2	11 1/2	12 3/4	13 3/4	15	16 1/4	17 1/2	18 1/2	19 1/2
Depth of groove....	h	1/8	1/8	1/8	1/8	1/8	1/8	1/8	1/8	1/8	1/8	3/16	3/16	3/16	3/16	3/16	3/16	3/16	3/16	3/16	3/16
Inside diameter of tongue ...	H	1 13/16	2 5/16	2 13/16	3 5/16	3 15/16	4 7/16	4 15/16	5 7/16	5 15/16	7 3/16	8 3/16	9 3/16	10 3/16	11 11/16	12 11/16	13 11/16	14 15/16	16 3/16	17 3/16	18 3/16
Outside diameter of tongue ...	J	2 9/16	3 1/16	3 9/16	4 1/16	4 11/16	5 5/16	5 13/16	6 15/16	6 13/16	8 1/16	9 3/16	10 7/16	11 7/16	12 11/16	13 11/16	14 15/16	16 3/16	17 7/16	18 7/16	19 7/16
Height of tongue ...	i	3/16	3/16	3/16	3/16	3/16	3/16	3/16	3/16	3/16	3/16	1/4	1/4	1/4	1/4	1/4	1/4	1/4	1/4	1/4	1/4
Height of tongue in lens joint.	j	9/32	9/32	9/62	9/32	9/32	9/32	9/32	9/32	9/32	9/32	3/8	3/8	3/8	3/8	3/8	3/8	3/8	3/8	3/8	3/8
Diameter of gasket in lens joint.		7/32	7/32	7/32	7/32	7/32	1/4	1/4	1/4	1/4	1/4	5/16	5/16	5/16	5/16	5/16	5/16	5/16	5/16	5/16	5/16

This table is abridged from tables of standard pipe fittings for pressure up to 20 atmospheres (294 lbs.), adopted by the "Verein Deutscher Ingenieure," in the year 1900. All the dimensions, originally given in meters, have been changed to read in inches and fractional parts of an inch, and have been slightly increased over what the converted figures would be. Number and size of bolts have not been changed.

Fig. 15. High-Pressure Steam-Pipe Flanges

on the continent of Europe to construct elbows, T's, water-catchers, steam collectors, etc., also with loose flanges. Indeed, loose flanges are more universally used in Germany than in any other country, and it is therefore to be expected that one will find there a larger variation in their design than in America. As previously stated, loose flanges are designed for both wrought-iron and steel pipes; the welding process, being no longer secret, is performed in many different shops with a consequent large variety of loose flange joints.

In America the ground joint is much more prevalent than in Europe, where the tongue-and-groove joint is much favored, the gasket usually consisting, for high-pressure superheated steam, of corrugated copper rings sometimes imbedded in asbestos; also, corrugated steel may be used, the latter being the practice to a certain extent in the new subway plant in New York. In the flange already referred to, Fig. 15, at the right-hand side a so-called lens joint is illustrated, which is somewhat similar to the above-mentioned Riley joint, but the ends of these pipe sections are made considerably broader, and it will be noticed that a round gasket is used, usually consisting of a woven-copper ring. By using reinforced pipe ends, as is the custom with Continental engineers, any type of tongue-and-groove joint may be adopted, and it will depend altogether upon the pipe for its contact surface, and not upon the loose flanges. The reinforced end is of the same material as the pipe itself, previously forged as a short cylinder with a heavy flange, and then welded to the pipe, after which the joints are turned and finished.

The loose-ring flange is made of wrought iron, cast or forged steel. It will be noticed that between the nut and pipe body sufficient space is left, although the outside diameter of the flange is less than that common in America for high-pressure steam flanges. Of course, the arrangement of the flange with relation to the pipe itself varies.

Figure 16 represents a flange movable on a reinforced

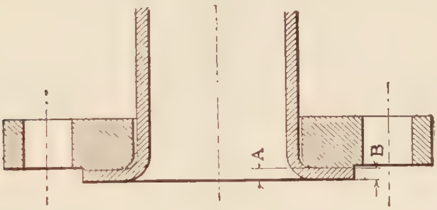


FIG. 16

pipe, the abutting faces of which are flat, while that in the table are at 45 degrees. When simply flaring the pipe ends, as shown in Fig. 16, the pipe becomes thinner at *B* than at *A*. Of course, it, therefore, becomes necessary to give the abutting face of the loose flange a slight slope, in order to obtain a complete contact. If this is not done the joint will sooner or later leak, and in any case the pipe flanges are thinner than the straight-pipe shell. To overcome this, a flange similar to that illustrated in Fig. 17 may be adopted, which has been successfully used on the continent of Europe. This type is not simply flared over, but during the process of flaring it is upset and the flange made somewhat thicker than the pipe shell, thus making it

possible to mill the faces without reducing the thickness below that of the pipe proper. This type has been

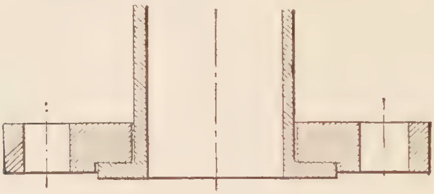


FIG. 17

much used on the continent of Europe and is to-day regarded as an effective pipe joint. Fig. 18 represents

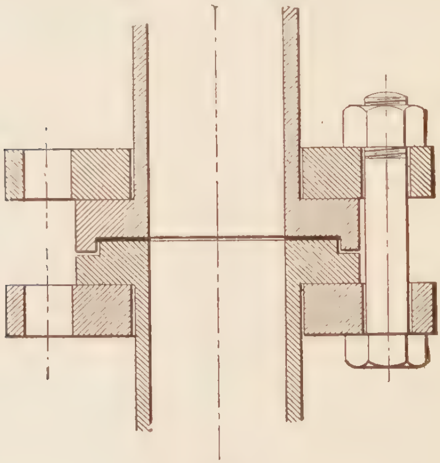


FIG. 18

a flange joint of male and female type, and it may as well be made with tongue and groove, or smooth face.

This flange is welded to the pipe, and the ends then faced. Figs. 19 and 20 represent loose-flange joints with the pipe rolled in the flange. In both cases only one of the flanges is of the loose type, while the oppo-

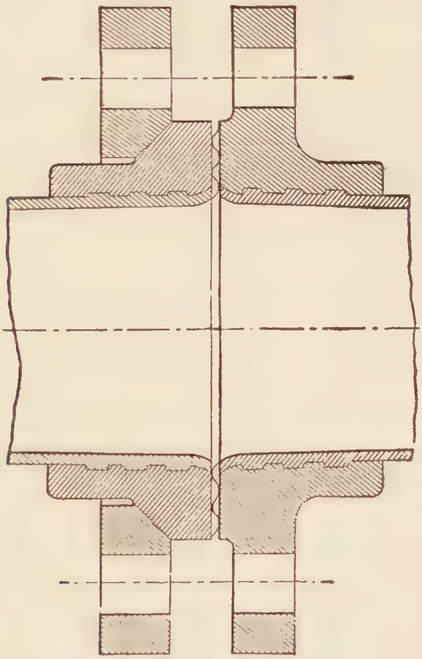


FIG. 19

site one is fixed to the pipe. A screwed joint, also of the loose-flange type, is illustrated in Fig. 21. As the screw fittings are not so favorably considered in Europe as in America, this type is seldom found,

especially with high-pressure steam. It is not so much the objection to the screw joint on account of its tightness, but to the liability of rupture at the end of the threaded section of pipe, especially in the case

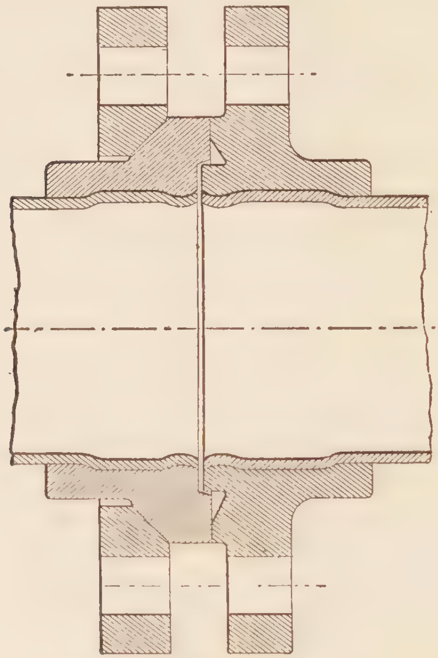


FIG. 20

where the pipe is not upset and the expansion and contraction work at this weak spot. Another type of flange is shown in Fig. 22, where the flanges are riveted to the pipe sections, and although the pipe joints, as

well as the rivets, are well calked, it will easily be seen that each individual rivet increases the liability to leakage. Where a flange has to be made in the power-house itself, a flange similar to that illustrated in

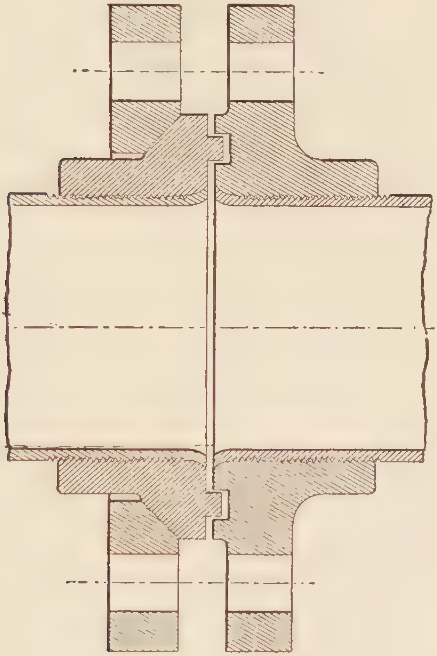


FIG. 21

Fig. 23 may be advantageously employed. This type is patented by the Sulzer Company, and is used for filling in sections, generally between two cylinders, or pipes carrying low-temperature steam. It will be seen

that, as the steam on leaving the high-pressure cylinder is not longer superheated, the temperature does not interfere with the use of copper. The boiler feed-water pipes, in America, are often made of brass with screw flanges, while on the continent of Europe these pipes are usually of copper.

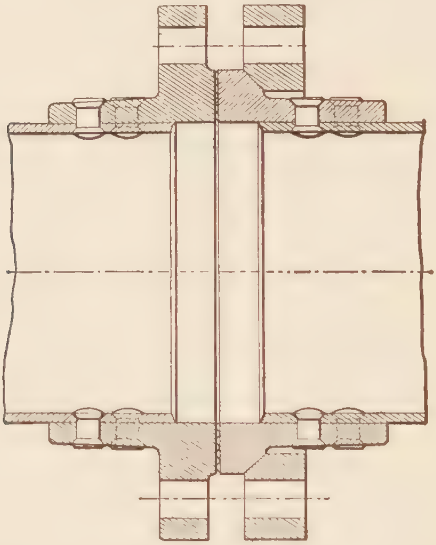


FIG. 22

Figures 24 and 25 show a joint which is preferable to all others for exceedingly high pressures, both for superheated steam and hydraulic work. This joint is made for superheated steam by inserting a plain gasket of annealed copper between the serrated faces, as shown at Fig. 24, and pulling together Fig. 25. The bolts in

FIG. 23

all of these flanges are made of sufficient strength to compress the copper into the desired form. The amount of surface on this gasket is of such proportion

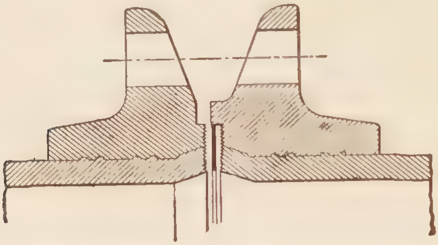


FIG. 24

that the pull on the bolts is sufficient to obtain the desired results. For high-pressure hydraulic work, with pressures up to 6000 pounds per square inch, a lead gasket has been used. The pressure this joint

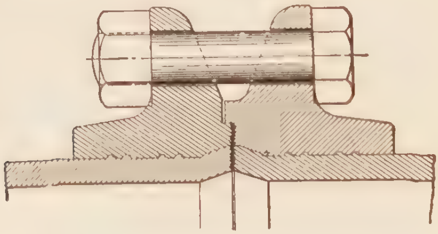


FIG. 25

will stand is limited only by the strength of the flange and the bolts. When made in proper proportion, a pressure of 20,000 pounds per square inch could be

carried as easily as that of 6000 pounds, as it is absolutely impossible to have a leak when properly made.

Figure 26 shows a joint similar to Figs. 24 and 25, with the exception that the gasket and serrated faces are omitted and the face of the pipe and flange is so finished as to form a flange-to-flange joint. This need not necessarily be a ground joint, inasmuch as the bolts are so proportioned as to give a pressure of over 1000 pounds per square inch of surface contact. This

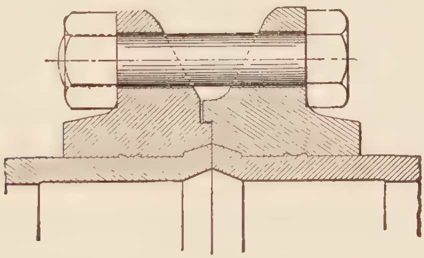


FIG. 26

joint is preferred by many engineers on account of there being no possibility of its being affected by expansion and contraction. It is also impossible for the joint to leak between the flange and the pipe when it is properly attached. This form has been tested up to 3500 pounds per square inch with the flange loose on the pipe; this, however, was in a special case, wherein the pipe was subjected to an enormous pressure sufficient to cause the flange to be strained beyond the elastic limit, and thus give the flange a permanent set, causing it to become loose on the pipe.

After finding out this particular feature, it was thought advisable to see what the joint would stand without leaking between the flange and the pipe, and as before

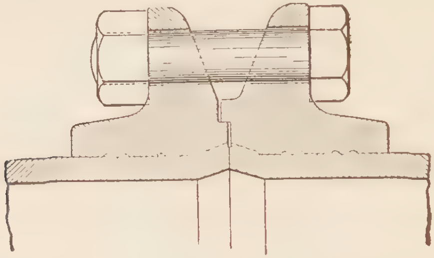


FIG. 27

mentioned, it stood 3500 pounds per square inch, with no sign of leak. This joint possesses many advantages over other forms wherein the metal is turned over at the ends of the pipes and faced so as to form a metal joint.

Figure 27 shows a modification of Fig. 26, wherein

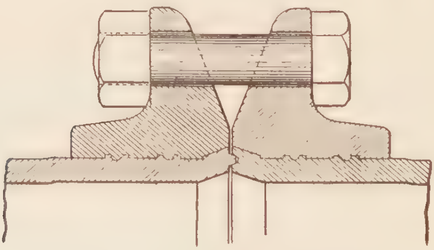


FIG. 28

the entire joint is made at the ends of the pipes, the flange itself being cut away clear, as shown.

Figures 28 and 29 show a new form of joint, as

suggested by Robert S. Riley. This joint is considered by many to be of great value for superheated steam only, and is what we may term a metal-to-metal joint, having the ends of the pipes abutting and the flanges clear.

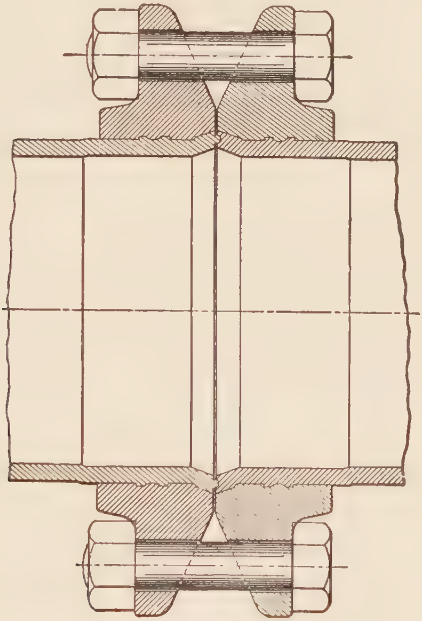


FIG. 29

Figure 30 shows another form of joint which is practically the same as Fig. 26, with the exception that it has an additional groove in the flange, and a gasket shown between the faces of the flange and pipe. This joint will stand any known pressure if properly proportioned.

Figure 31 shows a joint similar in character to Fig. 25, with the exception that the male and female at the ends of the pipe are dispensed with, and the pipes and

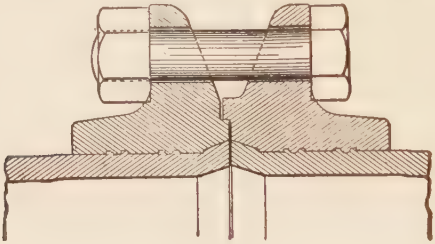


FIG. 30

flanges are formed as shown, with the gasket between them, or they may have a metal-to-metal joint as in the previous cases referred to. In fact, any of the before-mentioned types of joints may be utilized without the male and female feature if desired. This

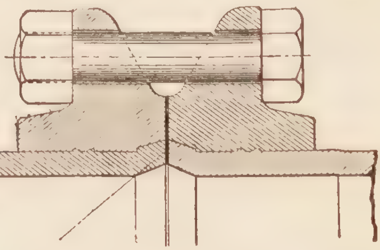


FIG. 31

last-mentioned joint possesses all the good features possible in any metal-to-metal joint, with the exception of the flange being fixed instead of loose. This,

however, can easily be remedied by using the Allen joint, previously referred to, and making the faces plain instead of male and female. The strength of a joint made on this principle is unlimited, and depends only on the designer. Five thousand pounds pressure has been placed on a joint designed for testing to this pressure, but intended for use on steam-pipe lines carrying 300 pounds pressure.

In all of the designs referred to, they can be applied to flanges ranging in sizes from 2 up to 20 inches in diameter, and every flange has been so calculated as to produce a definite pressure on the face or gasket, a suitable number of bolts of ample size having been provided to produce this effect. The thickness of the flange in every case has been calculated as a beam supported at one end and loaded at the center of the bolt, and the breadth of the beam considered as the distance between centers of bolts, thus giving a uniform design throughout the entire list of flanges, wherein the bolts and flanges are equal in strength.

This new list is suitable for working pressures between 100 and 300 pounds per square inch. The male and female may be omitted on pressures below 150 pounds.

Another object attained in designing flanges here shown is to keep all bolts outside of the joints as will be seen, so as to render them free from corrosion, and also avoid the troublesome effect so often encountered of flanges leaking around the bolt-holes.

V

PACKING FLANGE JOINTS WITH SOFT PACKING *

After the packing in a flange joint has blown out (to use a common expression which usually means that a small part of it has failed), the next move is of course to remove the bolts holding the flanges together. When nuts have been on bolts in such a place for several years it is often easier to twist one or more of them off than to remove the nuts in good order. If other bolts are at hand to replace those spoiled it does not pay to spend much time trying to save them; but many times there are no more in stock, hence it is desirable to save all of the old ones. Kerosene oil is one of the best things known for loosening a rusty nut, but it requires time for it to work, and as it is not always practicable to wait, some other means must be adopted.

A very good plan is to hold a sledge or heavy hammer against one side of a nut and strike the opposite side several smart blows with a lighter hammer, as this will usually loosen the rust and enable the engineer to remove the nut without further trouble.

Having taken out all bolts, put them into a pan

* Contributed to Power by W. H. Wakeman.

containing kerosene oil and let them remain there until wanted for use, for by so doing the rust and dirt will be loosened.

Flanges do not always come apart readily after the bolts are removed, therefore chisels or wedges must be used to force them apart, for some kinds of packing adhere firmly to iron heated by steam. Do not attempt to separate them in such a case by one wedge only, as the heavy strain brought to bear at one point may break the flange. It is better to use two or three in a stubborn case than to run any risk in the matter.

If the stop valve near the boiler is tight, preventing escape of steam, it will be appreciated at this time. A small leak may not cause a postponement, but in some cases it is necessary to remove all steam from the boiler before the job can be finished, and this means several hours' delay, besides a loss of much heat. Have the stop valve made tight, or put in a new one as soon as possible.

Care should be taken to remove all of the old packing, and for this purpose a carpenter's chisel may be used, as its shape is well adapted to the work, for it can be used where the flanges are but a short distance apart. The carpenter may object, but in an emergency it is better to buy him a new one than to delay the work.

Where a scraper is wanted it may be made by turning the end of a file at right angles to the body and grinding it to a sharp edge of suitable form.

A few lines concerning selection of packing for flanges will not be out of place at this point. In some of our old plants cast-iron pipe is still in use.

This pipe has been used just as it came from the foundry, and of course the face of every flange was very rough. In such a case it is necessary to use a soft fibrous packing, not less than $\frac{1}{8}$ inch thick, or it may be well to use one size thicker, in order to fill all depressions before the high spots come together.

The use of thick packing is to be avoided as much as possible, because it is more expensive in first cost and more liable to blow out, all other conditions being equal, thus increasing the cost of maintenance. Common rubber packing with cloth inserted is of no value whatever, when compared with improved brands made on purpose for such service, without cloth or canvas. There are several good kinds on the market at the present time. If steam passing through the pipe is saturated with cylinder oil, it will gradually dissolve soft packing, but asbestos millboard stands this test well. If used on rough flanges it should be at least $\frac{1}{4}$ inch thick, and even then it may be difficult to make a tight joint. On smooth surfaces $\frac{1}{16}$ inch is sufficient.

Where it is found difficult to keep a joint packed with either of these, some special kind may be used to better advantage, but if put in properly the above-mentioned brands will last for years, unless there is some defect in the piping, which should be corrected without delay, as it may be dangerous.

The next point to be considered is the form of gasket to be used. If it is decided to cover the whole face of flange with packing as shown in Fig. 32, and it is practicable to widely separate the flanges, the packing may be spread over one of them and by

hammering it over the edges of the iron it will be cut through, making an acceptable fit. By striking directly over the holes with a ball peen-hammer, the bolt-holes can be nicely cut in a short time.

However, in many cases it is not possible to widely separate the surfaces to be packed, therefore it becomes necessary to ascertain the outside and inside diameter of the flange. Then, by using a pair of dividers, the

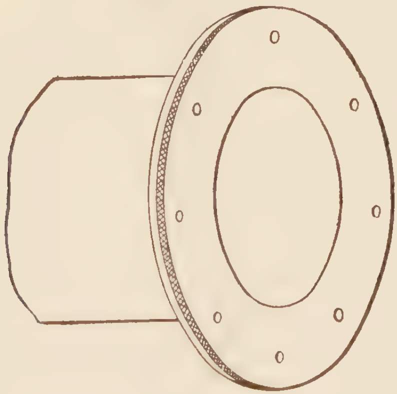


FIG. 32

gasket can be laid out, and by following these lines with a sharp knife, a neat job is the result.

There are two points in favor of using a wide gasket, one of which is that when placed in position between two flanges that are close together, it is an easy matter to bring it into line by observing the outer edges and locating them flush with the outside of flanges. The other point is that when the nuts are screwed on, it is

practically impossible to make a mistake, in doing the work, as may happen when a narrow gasket is used.

The objection to this form is, however, that on account of its great width it is liable to bear at some undesirable point rather than where it will do the most good. This will be apparent by considering the narrow gasket illustrated in Fig. 33. Its internal diameter corresponds to nearly the inside size of flange, but it

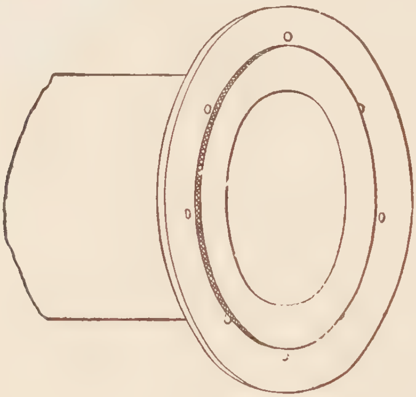


FIG. 33

is only wide enough so that when placed in position it is wholly within the bolt-holes. The result of this is that all pressure caused by screwing up the nuts is concentrated on a comparatively small surface, therefore the pressure per square inch is much greater than where a wide gasket is used, because in the latter case the same total pressure is distributed over about three times as much surface.

With a wide gasket the nuts may be screwed up according to any convenient plan, but with a narrow gasket care must be taken to tighten them as nearly even as possible. If this caution is ignored and one nut tightened or screwed down as far as possible before others are touched, the result will be as in Fig. 34.

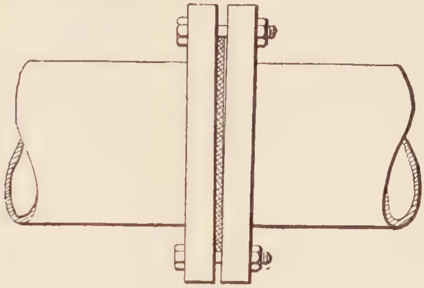


FIG. 34

This is exaggerated in order to make it plain, but in practice the flanges bind on the gasket at one point only, consequently even a light pressure of steam will blow out the opposite side of it, and make it necessary to put in a new one. In one case less than 5 pounds did this, requiring a new gasket to make good the damage.

One nut should be screwed down until a light pressure is brought to bear on the gasket, then the nut directly opposite should be treated in the same way. Another should be treated in like manner, followed by the nut directly opposite to it, and so on, until all of the bolts are under light tension. The same process should be carefully repeated, giving all bolts more

tension. Going over them once more is sufficient to finish the job and hold the gasket at all points evenly.

Perhaps some reader will laugh at the idea that a flange joint could by any possibility be packed without a hole through the center of the packing, but cases are known where the circular piece was left whole and much surprise manifested when nothing could be forced through the pipe.

It appears as if pressure would rupture the packing and it undoubtedly could not hold on a large pipe, but for the smaller sizes (say, 8 inches or less) it will withstand more than anybody naturally expects. Always cut this center hole a trifle larger than the internal diameter of the flange.

When laying out a gasket, locate it as near to the edge of the sheet as possible, in order to save packing. The circular piece left after cutting out a large gasket may be used for a smaller size until what is finally left is only large enough for packing a union.

After a joint is nicely packed, steam should be admitted to the pipe slowly in order to warm it gradually, for sudden application of pressure, which may be air pressure in advance of the steam, may blow out the gasket. After a light pressure of steam has been on long enough to thoroughly warm the joint, it is a good plan to remove the pressure and tighten all nuts holding the flanges, as is it often possible to imbed the flanges more firmly into the gasket after both are well heated.

The practice of screwing up nuts when pressure is on the gasket cannot be too strongly condemned, as n

is dangerous. If a joint is loose and one nut is turned without removing pressure from the pipe, an overload is at once brought to bear on this bolt, and there is a good chance for it to fail under the tension. The failure of one bolt immediately throws an extra load on the others, and if they do not fail under the abuse it is due to good luck more than to good management. Of course it may not be convenient to remove pressure from the pipe when a gasket begins to leak, but in that case it is better to let it leak or even blow out than to run the risk of sudden failure of the bolts under tension.

IV

CONNECTING BOILERS TO STEAM MAINS *

PROBABLY no detail connected with the erection and operation of boilers receives less attention than the manner of connecting the boiler to the main steam line unless it be the method of putting the boiler into commission after it has been laid off for cleaning and repairs.

A number of accidents caused by the improper design of the branch piping are illustrated in this chapter by sketches, which sketches are shown by Figs. 35 to 43 inclusive. These sketches certainly show a variety of arrangements and sometimes two or three types are found in the same boiler-house.

The first thing to be determined regarding the connection between the boiler and the main should be the number and kind of valves to be used. The most common practice is to use one globe valve, but the only argument in favor of this is the lower first cost. It is better practice to use two valves in this branch for considerations of safety and convenience. Surely no one should be placed in jeopardy of such a horrible death as scalding in a boiler, for the sake of the first cost of an extra stop-valve. Then boilers have been

* Contributed to Power by W. E. Snyder, M.E.

known to go for months in need of inspection and repair in the drums, yet the work could not be done, nor could any cleaning be done because the crown valve was leaking and no one could go into the drums. Then, again, it is not a pleasing sensation to feel, while in a boiler, that there is only one valve of a very uncertain design and construction, and of unknown age between you and a horrible death. Nor is the single valve with its usual leaks calculated to inspire the workman with the sense of complete security so vitally necessary if good work is to be done in a boiler. Therefore, two valves should be placed in each branch. If the boiler is to be off but a short time and no one is to go into it, only one need be shut; but when internal cleaning and repairs are to be done both valves can be closed and the greatest degree of safety secured.

As to the kind of valves to be used, a first-class gate-valve is the most desirable, though globe valves are probably more frequently used. One reason for preferring the gate to the globe valve is that the gate will open more gradually and this feature may at some time be the means of saving some poor fellow's life.

Another reason for preferring the gate-valve is that there is no possibility of a careless or ignorant workman putting it in so that, when shut, the pressure is against the stem tending to strip the threads, as has been done with globe valves.

The objection to the use of angle-valves, however, is much stronger than to globe valves, first because they have the same rapidity of opening as the globe valve, and, second, especially because the direction of flow

must be changed in the valve and it is therefore a form of construction peculiarly liable to injury by water ram if there be any chance for water to collect in the pipe. Examples of this will be given below.

In Fig. 35 the water tender had been accustomed to

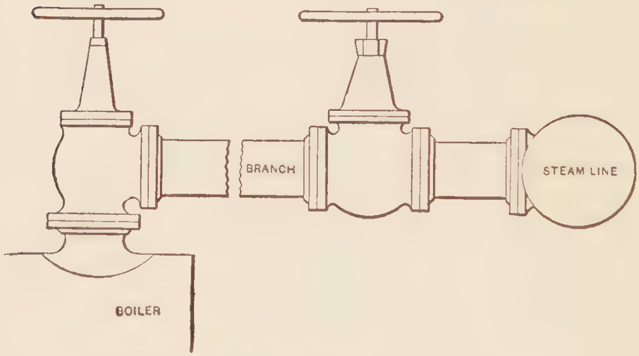


FIG. 35

close both valves shown while working in the boiler, and then, before getting up steam, the angle crown valve would be opened, and when the boiler was ready to turn into the line the globe valve in the branch would be opened. The branch inclined slightly toward the boiler.

One day, owing to some misunderstanding on the part of an attendant, the crown valve was left shut until after steam was up. The globe in the branch had been left open as no one had been in the boiler. The water tender was an intelligent man and studied over the matter for some time as to whether he would

let steam go down again and go through the regular routine with the valves, or risk opening the crown valve when steam was up. He finally decided on the latter course, but at the first movement of the hand wheel on the angle-valve, the most violent pounding began inside the valve and seemed to resemble the blows of a heavy sledge. The water tender was terribly frightened, but seized the wheel and whirled the valve open as rapidly as possible. The pounding gradually subsided, leaving the man so badly frightened he could hardly get down off the boiler.

It is reasonably certain this was due entirely to water ram caused by collection of water in the branch, and the only reason the valve was not burst was either because there was not enough water pressure to do serious damage or the valve was an unusually strong one.

The next instance of similar trouble was with the connection shown in Fig. 36, where one angle-valve alone was used. The boiler had been off for some time while the rest of the plant had been running and of course considerable water collected in the branch pipes as the drip had been left closed.

When the boiler tender went to put the boiler on the line he found the valve quite cold, though there was 125 pounds steam pressure on the system. Thinking to drain out the collected water, be began to open the small drip-pipe valve. At the first turn a violent surge occurred in the branch pipe followed by the bursting of the angle-valve on the broken line shown. No further damage followed, as it was possible to cut

the other boilers off this part of the steam main, and this was done at once.

The next accident of a similar nature was caused by the connection shown in Fig. 37. There were two valves in this branch: a globe valve in the vertical leg and an angle crown valve. The boiler had been off

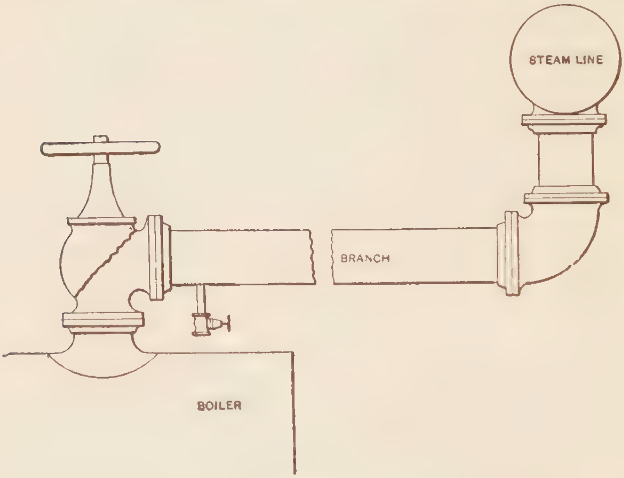


FIG. 36

for some time, the remainder of the plant being still in operation. The globe valve in the vertical leg had been left open and only the angle-valve had been used to shut the boiler off the line.

When the boiler was ready to be put on the line again the water tender went up on the boiler and began to open the angle-valve. At the first movement of the wheel the valve burst, as shown in the figure.

The next accident of this kind to be cited was caused by a connection shown in Fig. 38. In the connection a gate-valve was placed next to the boiler and from this the branch pipe ran about 30 feet to an angle-valve which connected it to the steam main.

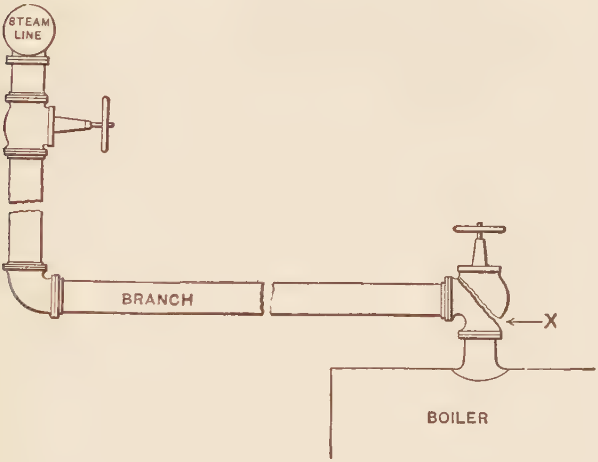


FIG. 37

It had been customary to raise the pressure to about half the working pressure and then open this gate-valve. The pressure would then be run up until the boiler was ready to put on the line, when the angle-valve would be opened. One day the gate-valve was opened very shortly before the boiler was to be put on the line. The water tender then went to open the angle-valve and while doing so stood with his left foot on the steam main and his right up on the branch

pipe, thus placing him at the side of the angle-valve. At the first movement of the valve there was a sharp click and an instant later the valve burst as shown by the ragged line. The water tender was knocked from the pipe, fell to a platform about 5 or 6 feet below and was instantly killed. The steam was at 150 pounds pressure and the body of the valve very well constructed of phosphor bronze $\frac{7}{8}$ inch in thickness.

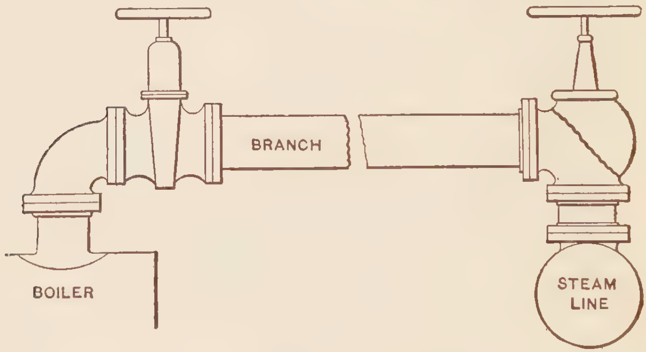


FIG. 38

All the circumstances connected with these accidents point to the collection or pocketing of water in the branch pipes. On one point, however, with reference to these accidents, satisfactory information is lacking and that is the relative steam pressures in the boiler and in the steam main when the valve began to be opened. The men who really had this knowledge, if any one had, were either in such a position or condition that they would not or could not say.

In questioning different boiler tenders on this subject,

the greatest variation was found in their practice. One man will positively assert that there should be several pounds excess pressure in the steam line while another will say the excess pressure should be in the boiler, and still another will say the pressures should be about equal.

Some of the particular points to be observed in the arrangement of the branches so as to provide as far as possible for safety are to avoid the use of angle-valves; to arrange the piping in such a way that condensed steam will not pocket in it, and to make provision for expansion. The danger of angle-valves has been shown above and other similar instances could be given. Some connections which have given no trouble are shown in Figs. 39 to 43, inclusive — the last two probably being the most satisfactory. These connections give little or no opportunity to pocket water and provide for expansion. Fig. 43 could be still further improved by placing another gate-valve between the bend and the boiler.

In case of any arrangement of piping in which water is apt to pocket, proper drip pipes should be placed where necessary and the water kept drained out.

Another important matter is to keep all pressure gages about a boiler plant properly adjusted so that they will show the correct pressure; then have them connected in such a way that it is possible to tell not only the line pressure but also the pressure in any individual boiler which may be off the line and in which steam is being raised.

Frequently so little care is given the gages that

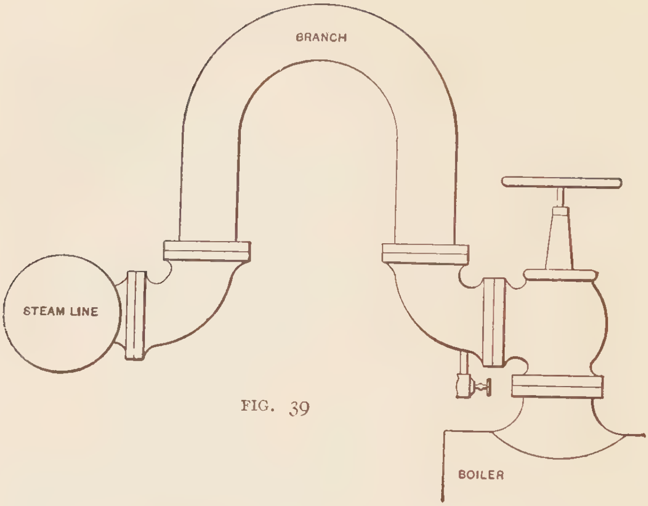


FIG. 39

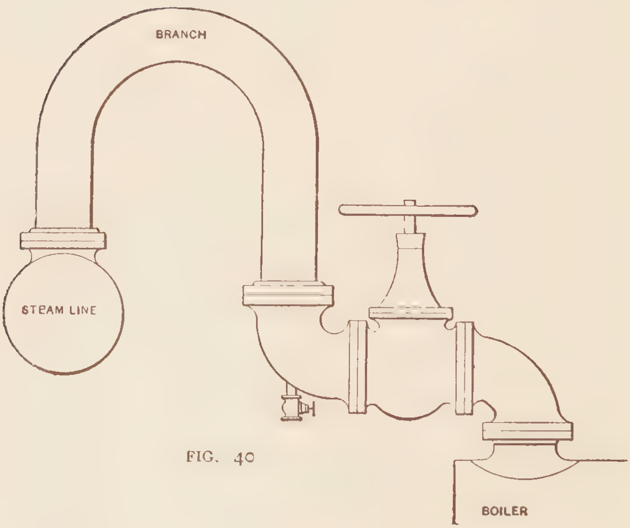


FIG. 40

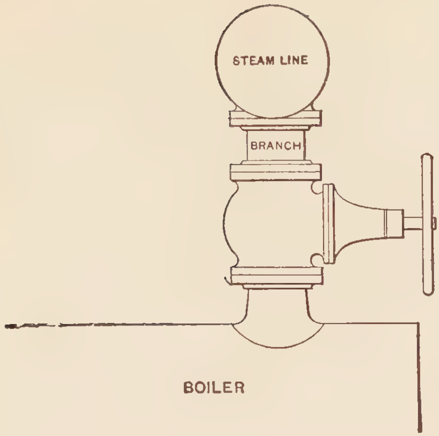


FIG. 41

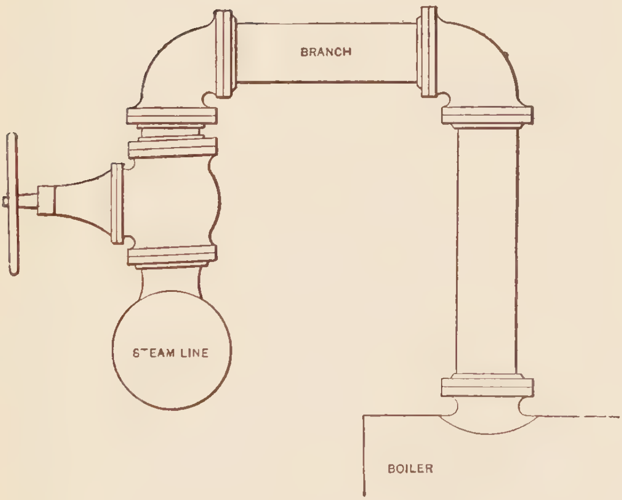


FIG. 42

there will be only one or two fairly correct for a dozen boilers and these are connected in such a way that they always show the line pressure. When steam is being raised in a boiler it is often a matter of pure

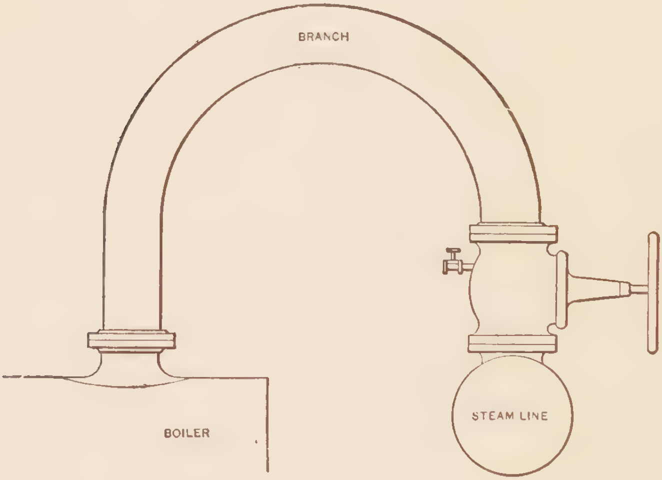


FIG. 43

guesswork on the part of the water tender to tell when the boiler is ready to turn into the line, as the boiler either has no individual gage on it or its gage is in such condition that it is practically useless. Inequality of pressure due to lack of or incorrect gages and pocketing of water in the branch pipes may cause such serious and costly accidents as described above.

VII

THE BURSTING STRENGTH OF STANDARD SCREWED CAST-IRON ELBOWS AND TEES *

It is a generally accepted fact that few, if any, of the accidents caused by the failure of pipe fittings are due only from the pressure of the gas or fluid contained in them, but are the result of combined stresses caused by the expansion and contraction of the pipe, water hammer or pipe improperly supported, any or all of these being in conjunction with a stress due to the pressure of the gas or fluid within the fitting.

A fitting may, therefore, be subjected to two classes of stresses; one, a legitimate stress, due to the pressure from that which the fitting contains, such as water, steam, compressed air, etc., and the other, due to strains caused by systems of piping improperly designed or installed. Stresses due to the former are tangible and may be estimated with some degree of accuracy, but stresses due to the latter are an unknown quantity varying with the wisdom of the designer and for which the factor of safety must provide.

The dimensions of fittings of different makers vary slightly, so that what is true of certain sizes of one maker's fittings might not be true of another's. The fittings upon which the series of experiments forming

* Contributed to Power by S. M. Chandler.

the basis of this thesis were made were those of a prominent valve and fittings manufacturing company.

In order that a fair average of the bursting strength of these fittings be obtained, three of each size were taken at random from a stock of pipe fittings, and the bursting pressures of the three averaged for a basis from which to determine the factor of safety. It was also thought advisable to cast a number of fittings of different sizes with one wall of the body thinner than in the standard, and to determine to what extent such a fitting was weakened and the factor of safety decreased. This was accomplished in the foundry by raising the core slightly, thereby adding to the thickness of the metal in the drag side by the amount taken away from the metal in the cope.

The result of all the tests made is shown in the accompanying table, and the strength of the weaker fittings, just mentioned, is shown in **bold-face** type.

All fittings were tested by applying water from a high-pressure steam-pump and measuring the pressure on a calibrated hydrostatic gage, the arrangement being shown in Fig. 44.

One or two incidents which came to light during the tests may be of interest. It was found, for instance, that a joint made up of red lead could not be made tight at the high pressures unless it had had a chance to get thoroughly dry; but that one made with tallow was tight as soon as made up. The tallow was melted and applied while fluid to the threads with a brush. On account of the flat surface exposed to pressure in a plug or bushing, it was found necessary in nearly all

tests to use solid plugs and reinforced bushings in order to prevent their failure before the bursting of the fitting to be tested.

In Fig. 45 are shown two curves which represent graphically the bursting pressures attained by the

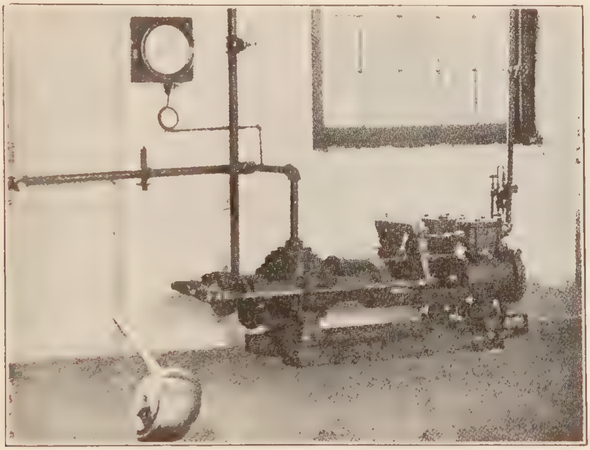


FIG. 44

elbows and tees, the curve *A B* representing the elbows and *C D* the tees, the different sizes being arbitrarily denoted on the horizontal scale, with the pressures represented vertically. The line of working pressure recommended by the manufacturer is shown below at 150 pounds and is taken as a basis for calculating the factor of safety.

It is seen that the elbows show a greater strength for all sizes than the tees, and reference to the photographs will show that failure of the tees occurred in a

majority of cases by breaking a piece from the body of the tee on one side where the metal has a nearly flat surface. This flat surface is entirely avoided in the design of the elbow and undoubtedly adds greatly to their strength. The fact that a tee has a larger inside diameter, measured through the run and the outlet, than the corresponding size elbow, also partially accounts for the higher pressure required to burst an elbow.

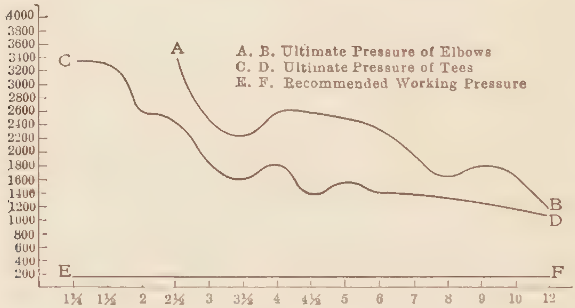


FIG. 45

The appearance of some of these fittings after fracture and the manner in which they burst may be seen in the reproduced photographs. The size and bursting pressure are given under each fitting.

The smaller elbow in Fig. 46 is one of 2½ inches, which did not burst at 3500 pounds pressure, the highest available with the apparatus. The size and pressures at which the various fittings failed as marked on each photograph and the methods of fracture as exhibited in the illustrations furnish an interesting study.

Fig. 47 shows the appearance of the T's after test.

FIG. 40

FIG. 47

STANDARD SCREWED CAST-IRON ELBOWS AND TEES—ULTIMATE TEST PRESSURES

Size	Elbows		Average	
$2\frac{1}{4}$	3500	3300	3400	3400
3	2400	2600	**2100**	2500
$3\frac{1}{2}$	2100	1700	2400	2250
4	2800	2500	2500	2600
$4\frac{1}{2}$	**2000**	2600	2600	2600
5	2600	2500	2500	2533
6	2600	2200	2300	2367
7	1800	2100	**1900**	1950
8	1700	1600	1700	1667
9	1800	1800	1900	1833
10	1800	1700	1600	1700
12	1100	1200	900	1150

Size	Tees		Average	
$1\frac{1}{4}$	3400	3300	3300	3333
$1\frac{1}{2}$	3400	3200	**2800**	3300
2	2500	2800	2500	2600
$2\frac{1}{2}$	2400	2100	2500	2450
3	**1400**	1900	1800	1850
$3\frac{1}{2}$	1200	1500	1800	1650
4	1800	2100	1700	1867
$4\frac{1}{2}$	**1100**	1400	1400	1400
5	1700	1300	1500	1600
6	1400	1500	**1100**	1450
7	1400	1400	1500	1433
8	**1200**	1400	1300	1350
9	1300	1400	1200	1300
10	1100	1300	1200	1200
12	1100	1000	1100	1067

VIII

BURSTING STRENGTH OF MALLEABLE-IRON PIPE FITTINGS *

A NUMBER of tests at the laboratories of a prominent valve and fittings manufacturing company, to determine the bursting strength of 4-inch standard, screwed, malleable-iron tees, black and galvanized, were made by S. M. Chandler. Following are given the ultimate test pressures at which these fittings failed, together with the weights of fittings tested:

The bodies of all fittings were $\frac{1}{4}$ inch in thickness.

Test No.	Black or Galvanized	Ultimate Test Pressure	Weight, Pounds
1	Black	2800 lb.	10.06
2	Black	3100 lb.	10.12
3	Black	2800 lb.	10.19
4	Galvanized	2700 lb.	10.62
5	Galvanized	3000 lb.	10.50
6	Galvanized	2800 lb.	10.75

The average bursting pressure of the black fittings was 2900 pounds, and of the galvanized fittings 2833

* Contributed to Power by S. M. Chandler.

pounds. As these tees were recommended for a working pressure of only 150 pounds, they therefore had a factor of safety of 19.3 for the black fittings and 18.9 for those galvanized, with a general average factor of safety of 19.1 for black and galvanized together. This is ample where the fittings are used at pressures recommended by the manufacturer and would even allow of safe usage at still higher pressures.

It is interesting to note that the galvanized fittings failed at practically the same pressures as the black. This is contrary to the expressed belief of many users of malleable fittings, who were very positive in their statements that galvanizing greatly weakened the strength of pipe fittings. Adherents to this theory claimed that dipping malleable fittings into a bath of molten zinc, and then suddenly cooling by immersing them in cold water, had a tendency to make the castings hard and brittle, bringing them back to the unannealed state. That such a theory is false is shown conclusively by the above tests, leading one to infer that the temperature to which castings are raised for galvanizing is not sufficiently high to injure them when suddenly cooled, providing, of course, that the castings have been properly annealed in the first place.

A comparison of the bursting strength of cast-iron and malleable-iron fittings as used commercially can be obtained by referring to a series of tests, in the preceding chapter. Here it was shown that three 4-inch standard, screwed, cast-iron tees were tested to destruction, failing at an average bursting pressure of 1867 pounds. As the average bursting pressure of

the six 4-inch malleable fittings referred to above was 2867 pounds, they were therefore about one and a half times as strong as those made from cast iron. While tests to support this statement were only made applying to 4-inch sizes, it is probable that the same ratio would be approximately correct for all standard sizes as the patterns for both the malleable-iron and the cast-iron fittings were made from one general design.

In most cases the malleable fittings developed leakage through minute "pin-holes" at pressures ranging from 1000 to 2500 pounds. In no instance were these pin-holes visible below a pressure of 1000 pounds with the black, or 2000 pounds with the galvanized fittings, while two of the galvanized fittings sustained pressures of 2500 pounds before pin-holes developed. It was therefore evident that the galvanizing was very effective in closing the pin-holes, which are generally characteristic of malleable fittings when used at high pressures.

A feature of interest that developed in these tests was the stretching of the metal in the fittings as the pressure increased. Careful measurement of the body of the tees with calipers before and after the application of pressure showed that the diameter had increased from $\frac{1}{8}$ to $\frac{1}{4}$ of an inch. This stretch caused excessive leakage in the threads of the fittings at the highest pressures and made considerable trouble and annoyance in conducting the tests. On this account it was found advisable to raise the pressure, in making the tests, as quickly as possible, as in so doing the fittings

were fractured before they had had time to stretch and leak to any great extent. This sudden application of pressure no doubt had a tendency to produce failure of the fittings at lower pressures than if the pressure had been applied gradually without shock. This difficulty could have been avoided by screwing the plugs farther into the tees when the pressure had reached a point slightly below that at which the fittings were expected to fail. Such a procedure would, however, be accompanied with danger, as any air entrained in the tee might cause a disastrous accident with probable loss of life in case of premature bursting of a fitting.

The accompanying sketch, Fig. 48, shows charac-

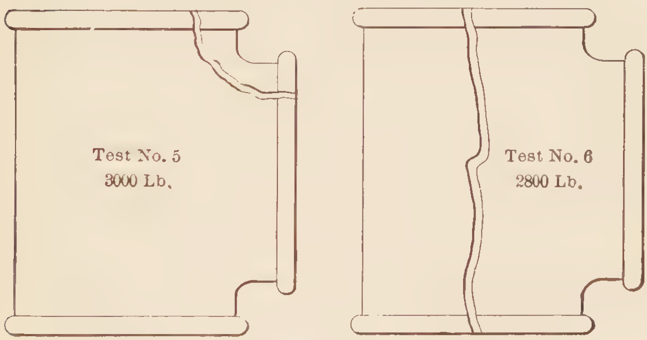


FIG. 48

teristic fractures of the fittings tested. Most of the tees failed as in Test No. 5, cracking through the beads at one end and the side outlet of the fitting. No pieces of metal were entirely separated from the tees when they failed, as was the case with the cast-iron

fittings. The malleable-iron fitting, being tough and slightly elastic, was simply torn apart while the cast-iron fitting, being more brittle, broke in many cases into separate and distinct pieces. These tests show plainly the superiority of malleable over cast iron for use in the manufacture of pipe fittings of small size.

IX

PIPING FOR A STEAM PLANT

THE accompanying sketch, Fig. 49, shows a piping arrangement which meets all the requirements of a small plant.

The piping is so designed that any one of the three units shown can be used for boiler feeds, or any two at the same time for any other purpose, such as in case of fire. In this case the two pumps can be used to force water on to the fire and the inspirator used as a boiler feed. This drawing may appear complicated, but a little study will simplify it. In case the pump P is used to keep the elevated tank T full of water the pump P' used to supply the boilers with water. By opening the valve B'''' water is drawn from the well and is forced to the Tank T by the pump P, or by closing valve B'''' and opening valve C' water is taken from the city main W to fill the tank T by the pump P. By opening valves C'''', D, D'', F'' and G, valves C, B''', being closed, water is taken from the well, forced through the heater and through the pipe S to the boiler. In case the pump P, as before, is used to keep the water up in the tank T and pump P' used in case of fire, the inspirator can be used to feed the boiler. Valves B''', S'' and C'' are closed, valves B''

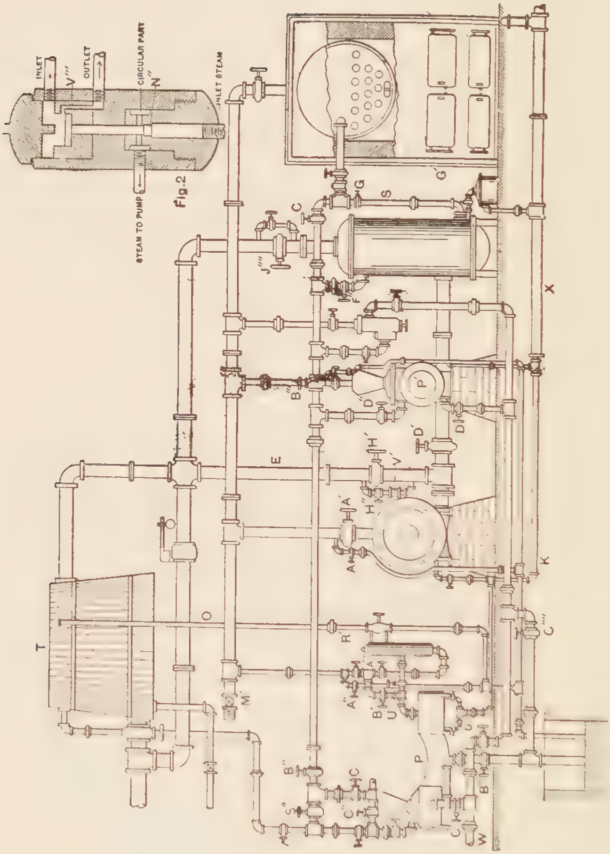


FIG. 49

C, D and C'''' are open, and pump P' forces water from the well through the nozzle N to the fire. With the valve B''' closed, water is taken from the well and forced into the boiler by the inspirator through the heater, or can be forced to the boiler and by-pass the heater by opening valve F'''', closing valves F'' and G. In case we should want to take water from the well for the tank T by pump P and city water for fire through pump P' and boiler feed through inspirator; by closing valves C' and C'''', opening valves B'''' and C''', water is taken from city main for fire by pump P' and boiler feed by the inspirator. It will be seen that by opening and closing of the different valves, the pump P or P' can be used to fill the tank or used in case of fire and still leave one unit for boiler feed. In some cases, as shown by the drawing, it is necessary to have a pump regulator; for instance, in laundries, where the water in the tank T is used for washing purposes. The tank T is open at the top, the exhaust steam after passing through the heater R is exhausted through bends or a coil placed in the tank T. This heats the water in many instances to the desired temperature. A regulator is placed on the steam-pipe, as shown at R', and works automatically. In starting this regulator first, valve A'' is opened and the water is all blown out of the steam-pipe shown. Then the valve A'' is closed and the valve A''' opened with the valve B' closed. The operation of the regulator is as follows: When the tank is partly empty or the water below the overflow pipe O, steam enters at the point marked "Inlet steam," in Fig. 2, under the end of the piston N'', raising it up.

By so doing, small ports are uncovered and steam is admitted to the circular part of the casting and from this point to the pipe marked "Steam to pump." When the tank *T* is filled to the proper hight, water will escape out through the pipe *O* on the tank to the pipe marked "Inlet," in Fig. 2. The weight of the water acting on this piston, with four times the area of the piston N'', forces the piston down against the steam pressure, closing the steam ports and stopping the pump. The water escapes through the port V''' to the pipe marked "Outlet" and back to the well, as shown. As soon as the water in the tank drops below the entrance of pipe *O* in the tank, the water escapes, reducing the pressure on the piston N''', the steam pressure raises the piston N'' and the pump is started up as before. In case the regulator gives out, the pump can be regulated by hand by the valve B' with the valves A''' and *U* closed. In starting the engine, the valve *A* is first opened and the steam-pipe is cleared of water. Valve *A* is then closed and valve A' opened, just enough to warm up the cylinder with cylinder cocks open, which in this case are connected at *K* to the drip or sewer pipe. After the cylinder is warmed up, the engine started and load on, the cylinder cocks are closed. It will be noticed in this drawing that the engine exhausts into the heater, but in case the heater cannot be used, by closing valves D' and J'''' after opening valve H', steam is exhausted up the pipe *E* to the tank *T*. When the back pressure exceeds what the back pressure valve is set for, it opens and the tank *T* is by-passed, thus relieving the pressure.

It will also be noticed that at the bottom of the heater at G', a trap is connected which keeps the heater clear of water of condensation. All drips or drains from the steam main and inspirator are piped to the well, and all drains from exhaust pipe drips from the cylinders are piped to the sewer or pipe X. At H' a small valve H'' is used to keep pipe E clear. A check-valve is placed in this small pipe at V', which opens toward the basement and closes upward. This allows the water to escape from the pipe E and prevents the exhaust steam from passing through it. When necessary, the little valve at J'''' can be opened to drain this pipe into the heater. The pipe at M'' can be used for anything that is needed. In this design all valves and unions are placed in the piping, so at any time any part can be taken down without interfering with the running of the plant. Again, it will be seen that only one valve is placed below the basement floor.

X

ACCIDENTS DUE TO FAULTY PIPING *

ACCIDENTS to steam-engines due to a sudden influx of water from steam, exhaust or drip pipes are so frequent that it may be interesting to review some of the causes due to defective construction, showing good and bad practice in piping to and from engines.

Boilers at times, for causes not always too obvious, prime, and greater or less quantities of water are carried over with the steam to the engine; this added to the condensation of the piping and cylinder not infrequently causes the wrecking of the engine.

Many engineers will point to the main steam line or header and say that they had it put in with a slant of three or four inches, so that all water will run back toward the boilers and therefore it is impossible to get water over into the engines.

This amounts to practically nothing against the current of steam through the pipe in the opposite direction, especially when a heavy load is thrown on, or in other words, just at a time when an accident due to water is liable to occur. Such a line will drain nicely when no steam is flowing.

In some cases we find large, long headers without a

* Contributed to Power by Thomas Hall.

sign of a drain and with steam lines rising from the top. In such cases, when the engines are running under light loads, water will collect in the header and when a sudden heavy load occurs it will be thrown over to the engine in slugs.

Reservoirs, separators, relief valves, explosion diaphragms, etc., are all good things, but none are absolute proof against accidents due to water in large quantities. In addition to such safeguards the greatest precautions should be taken in the design of the piping system. It is not at all a simple or easy matter to lay out a piping system with separators, etc., that will be accident-proof, in fact, few engineers are really capable of laying out a large piping system to the best advantage.

When an accident does occur it is often difficult to prove to the satisfaction of the plant owner the real cause, though it may be apparent to one experienced in such cases. The engineer in charge is not likely to give information which may reflect upon himself, and his disposition is rather to get on the defensive than to assist in locating the real cause. In other words, the tendency is to put it upon the engine builder, claiming some defect in workmanship, material or design.

The forcing of boilers beyond rating has a tendency to promote carrying over water. Some types are more liable to do this than others. Small steam-pipes are objectional because the necessarily high velocity of the steam materially assists in picking up the water and sweeping it over to the engines. The larger the

pipes or receivers, the slower the flow of steam, and consequently the greater the tendency for it to deposit the entrained water and vice versa.

The condensation in a large piping system is enormous, hence it becomes of very great importance to have the pipes thoroughly protected with a good non-conducting covering both for safety and economy.

The greatest care should be taken, when steam traps are used to drain steam lines to engines, to see that they are in the best working condition. These traps are placed in steam lines to perform certain work and when they fail to do this work they endanger all the machinery on the lines they are intended to drain.

Water glasses in separators should be considered as important as those that show the hight of water in the boilers. How many times one enters a plant and sees the gage glass missing on the separator.

Many accidents to engines can be traced directly to firemen allowing the water-level in the boiler to run too high. Sometimes this carelessness reaches the point of flooding the boiler, when the wrecking of the engine is almost sure to take place. At times, if the water is above its normal hight, a heavy load may be thrown on the engine, when the very rapid ebullition of the water, assisted by the high velocity of the steam, causes it to be thrown over into the piping system.

It is safe to assert that as many of the accidents which occur to engines are due to poorly constructed exhaust and drip-pipes as to any other cause. Several

sketches are here reproduced, showing bad treatment of both steam and exhaust, from actual cases in which the cause of the accident can be clearly traced.

Figure 50 shows an arrangement of piping where an

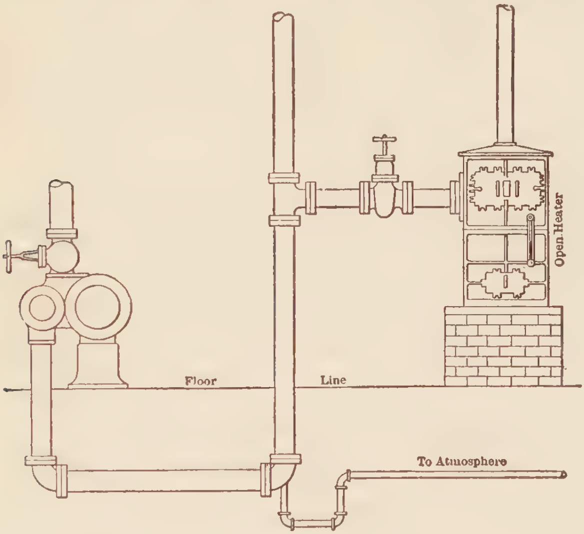


FIG. 50

accident occurred damaging an engine to the degree of an almost absolute wreck. The defect in the piping is at once apparent and shows how carelessly exhaust and steam piping at times are arranged.

The size of the drip-pipe was one inch, entirely too small for an 8-inch exhaust, and while this was supposed to take care of all the water coming from the engine, the outlet of this pipe was raised to a line

corresponding to the center line of the exhaust-pipe. This pipe should also have been tapped into the lowest part of the line instead of half way up the elbow.

Under these conditions satisfactory drainage was impossible. The exhaust-pipe filled with water, and on starting up was drawn into the cylinder, causing the breakdown.

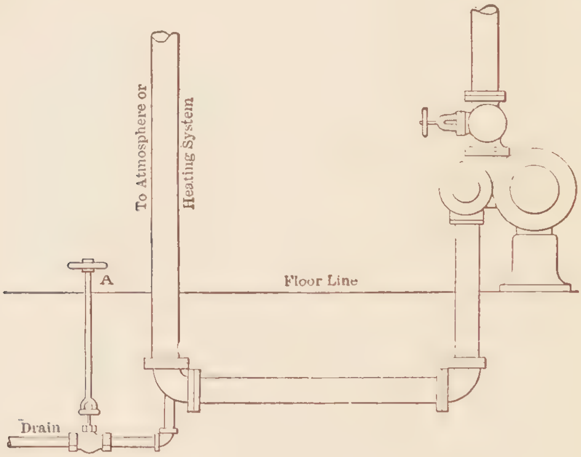


FIG. 51

Accidents have also happened with this arrangement of exhaust-pipe and open type feed-water heater, especially where the heater has no over-flow or other arrangement to prevent the water rising to the hight of the exhaust-pipe and flowing over into the engine.

Another manner of connecting the exhaust-pipe is shown by Fig. 51. This arrangement is very much

used, but if care is not taken in operating the valve *A*, damage to the engine is likely to result. Such drainage depends too much upon the memory and judgment of the engineer.

Figure 52 shows a case where an accident happened

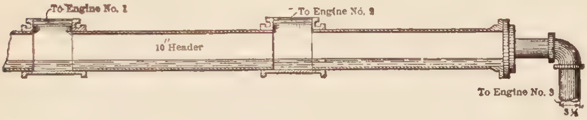


FIG. 52

to engine No. 3, which was of the slide-valve type. The cause was evident to the expert sent to investigate. He reported that while engines No. 1 and No. 2 were not running, the 10-inch header filled with water up to the level of the 3½-inch pipe, then when a heavy load was suddenly thrown onto engine No. 3, the water was carried over into it, causing the breakdown.

The operators of the plant were notified that unless a change in the piping was made the same accident was certain to occur again. Though they contemplated making the change, it was put off for a more convenient time. Exactly the same accident happened a second time some ten days later. The change was then made and no further trouble has occurred, although the engine has been in continuous operation over two years since that time.

The change included a drop leg draining the header by the boiler feed-pump, a separator above the engine throttle and a change of the 3½-inch pipe to the top instead of the end of the header.

An engine at the end of the line as this one was is always liable to get the worst of it.

Figure 53 explains itself and shows another erroneous

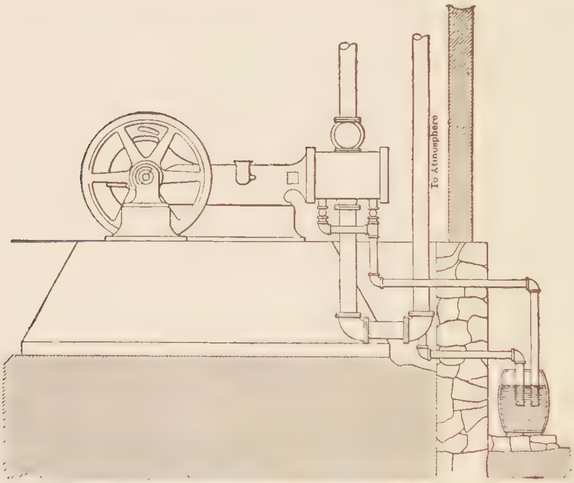


FIG. 53

method of terminating drip-pipes. The idea of this arrangement was to get the use of the warm water. Neither should exhaust drips be connected direct to sewers, as the sewer is likely to clog and seal the end of the exhaust-pipe with water, allowing the engine to lift it back into the cylinder in shutting down. This also applies to installations where drips are run too far into the hot well.

Instances have been known where the drips from the low-pressure cylinder of a condensing engine discharged

to the atmosphere instead of to the condenser. Care should be taken that these pipes do not connect to the condenser or exhaust-pipe at such a place as will make it possible for water to be drawn into the cylinder.

Figure 54 shows a line of piping by which steam is supplied to three engines. This system is frequently used with satisfactory results. The entrained water and that of condensation is carried with the steam and considerable quantities may pass over to the header.

Any large quantity of water, such as is occasioned by the priming of the boilers, is likely to be taken care of, due to the steam header construction, it being continued past the last engine and a drop leg, *A*, provided as large or preferably larger than the steam header and run down six to twelve feet and drained.

Figure 55 shows a good construction. The water leg should be made at least as large as the steam-pipe and about twelve diameters of the pipe in length. The boiler feed-pump furnishes a good drain in case the trap should fail to work. Instead of the drop leg a separator may be used, as in Fig. 56.

The arrangement shown in Fig. 55 will take care of a greater dose of water, though it is not so likely to take out entrained moisture.

A valve is a most unsatisfactory method for draining separators, drop legs, reservoirs or main headers. This method is dependent entirely upon the memory and judgment of the attendants. If opened sufficiently to insure thorough drainage, they are very wasteful of steam. It not infrequently happens that where such

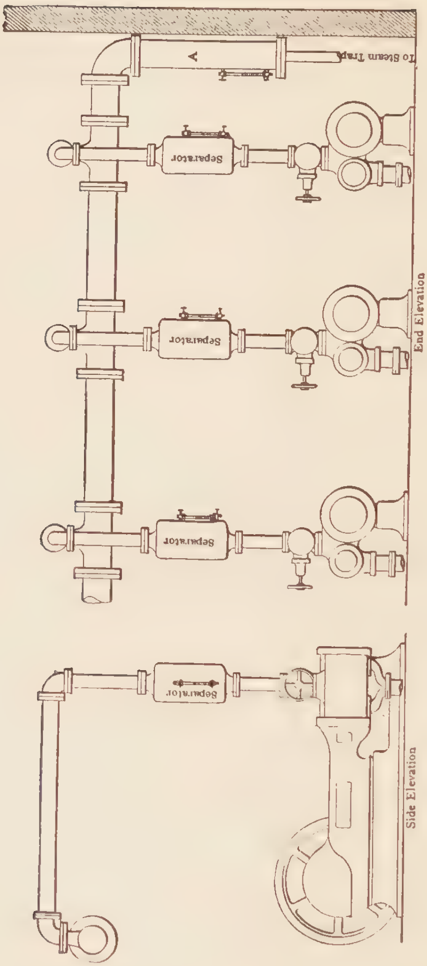
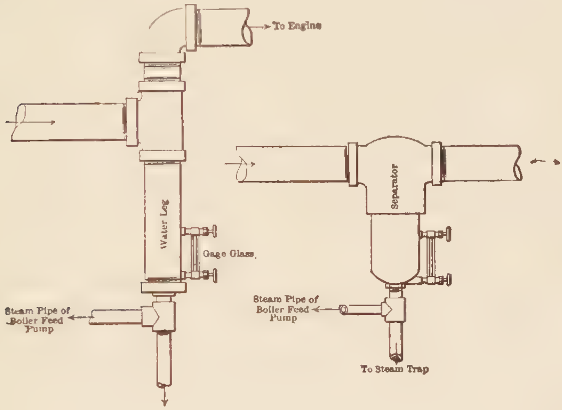

To Steam Trap
A
Separator
Separator
Separator
Separator
End Elevation
Side Elevation

FIG. 54

a method of drainage is in use, the engineer forgets to open the valve or he may not open it enough. Sometimes we see such drains with no gage glasses anywhere



FIGS. 55 and 56

except on the boiler. There had been, perhaps, one in the separator, but it was broken and not thought necessary to put it in again. What was the gage glass put there for if it was not necessary?

Starting and Stopping the Engine

Engines should under no conditions be started until they are thoroughly heated by blowing live steam through each end alternately and the steam-pipe and cylinder thoroughly drained of all water; drips should be left open until load is put on and then closed.

In shutting down it is preferable to leave drip-valves closed until the engine is stopped. If the throttle is

provided with a by-pass valve, it is well to close the throttle and stop the engine with the by-pass. By this means the engine can be brought gradually to a standstill, avoiding the pumping effect of the piston which occurs when the throttle is closed off entirely. If the throttle be closed quickly the momentum of the flywheel carries the piston back and forth for many strokes, causing it to act like a pump, so that any water with which the drips connect, or water which may be standing in the exhaust-pipe, is liable to be drawn into the cylinder and trapped when compression occurs.

Many accidents have been caused by simply warming the engine up at one end only. For instance, the engine is placed in a starting position at the cylinder head end and steam turned on. While the steam is heating the head end, water collects in the crank end and remains there. Then, upon starting the engine, this water is trapped in the crank end.

Frequently accidents happen to engines through confusion in condenser valves. The safest way is to start the condenser first, which frees the exhaust-pipe and drips of water; then close the valve in the exhaust pipe and start the engine non-condensing, exhausting through the relief to the atmosphere, throw on a partial load, and then open the main exhaust to the condenser, when the engine is ready for the full load.

In the case of high ratio and triple and quadruple expansion engines, however, it is often necessary to have the condenser in operation to get started. In such cases, presuming the engine to be warmed up and

drained, start the condenser, keeping the main exhaust valve between the engine and the condenser open. After the condenser has taken care of all water in the pipes and drains, start the engine slowly, giving it plenty of time to reach full speed. After the load is on and the engine is up to full speed, close the drains.

In stopping, keep the condenser in operation until the engine is shut down, then stop the condenser; this procedure will avoid possible accident.

The conclusions which may be drawn from the above are:

Avoid pockets in both steam and exhaust piping as far as possible. If these cannot be entirely avoided, see that efficient means for draining these pockets are provided. Do not use cracked valves for drains.

Use drop legs, reservoirs or separators or preferably a combination of either a drop leg or reservoir in the main header, with a separator above the throttle. Connect these up with efficient steam traps, or better still, as shown in Figs. 55 and 56.

Where a main header with several steam lines leading therefrom is used, always lead off these lines from the top of the header, and above all see that the header is effectively drained. It is advisable to use more than one drop leg in such a header.

Thoroughly cover all steam-pipes with a good non-conducting covering.

Keep the exhaust outlet below the cylinder if possible. If this cannot be done see that the pocket in this pipe is effectively drained, preferably by a free

open drain one-fourth the diameter of the exhaust-pipe. See that the cross pipe in a compound engine is effectively drained. All separators, steam reservoirs and water legs should have gage glasses, and these should be kept in operative condition.

If steam traps are used for drains make sure that they are operative.

To sum up, so design the system that a minimum of drains is required and make these automatic as far as possible. Do not add to the responsibilities of the engineer when it is unnecessary. Some systems seem to be designed with a view of making as many things for the engineer to look after as possible. It is little wonder that he should have an accident occasionally in such cases. A proper system is greater in first cost but entails less annual expense. Safeguards are a good insurance investment.

Do not connect the drips with drains or any other place where there is a possibility of stoppage and consequent flooding. Do not connect throttle drains, separator drains and engine drips together. Do not connect drains from two separators together; in fact, it is rarely advisable to connect up together any two drips into one line. It is better to run each independently to its own free outlet or trap, or run them independently to a large receiver tank, which must be effectively drained by a trap or other means. Where the boilers are located sufficiently far below the main steam header and separators to overcome any possible difference of pressure, these may connect to the boilers below the water-line, care being taken to see that

they are so connected that they will operate properly when one or more boilers are shut down.

Provide ample-sized steam-pipes, equip the engine with explosion diaphragms or relief valves. Warm the engine thoroughly before starting, with all drips partially open. Turn the engine over slowly some minutes before bringing up to full speed, then close drips. Shut down slowly, and close the drips after closing the throttle.

XI

PRACTICAL SUGGESTIONS *

THE writer was recently shown a simple and reliable device for ascertaining at a glance whether a steam trap is performing its duty properly.

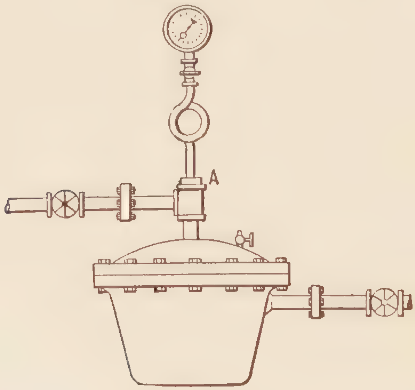


FIG. 57

The trap in question is attached to an apparatus, the efficient operation of which necessitates the immediate removal of all condensation. The failure of the trap to act would cause the loss of several hundred dollars' worth of stock.

As shown in Fig. 57, a tee is placed in the discharge

* Contributed to Power by C. W. Oakley.

pipe at *A* in place of the usual ell, and a steam-gage is connected at that point. As the trap discharges the gage indicates a rise in pressure according to the velocity with which the water is discharged.

The gage need not be an accurate or expensive one, as any old gage which has been discarded on account of inaccuracy will answer the purpose providing it will indicate a rise and fall of pressure.

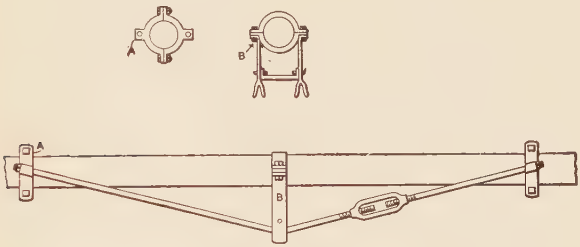


FIG. 58

Considerable difficulty is sometimes encountered in supporting long lengths of steam or water pipe, which cross yards, roads or other places where it is impossible or inadvisable to place hangers or posts for that purpose.

Figure 58 illustrates a method of overcoming the difficulty by providing a truss made of steel or iron rods on each side of the pipe, secured at the ends to suitable cast or wrought-iron clamps having lugs on each side to receive the rod ends, as shown at *A*, Fig. 58.

A collar or clamp is placed in the center of the span, from which risers extend to the truss rods, as at *B*,

Fig. 58, the lower ends of the risers being forked to receive the rods.

Care must be taken in the case of steam or hot water pipes not to draw the truss rods up tight until the maximum temperature is reached, otherwise the increased length of the pipe due to expansion will draw the rods too tight and cause the pipe to buckle in the middle.

One of the problems confronting the engineer in the installation of large vertical boilers where the lack of room compels the making of close connections, is that of properly supporting the steam piping so that there will be no undue strain on the fittings whether the boiler is under steam or lying idle.

Take the boiler in Fig. 59, for instance, where the expansion vertically when under 90 pounds steam pressure is $\frac{5}{8}$ inch. It will be readily seen that any non-compensating support which would properly carry the weight of the pipe and fittings when cold would be utterly useless when the boiler was in service. Likewise the same support adjusted to carry the piping when in use would subject the pipe and fittings to an enormous strain when the boiler cooled down and the hight decreased.

To overcome this difficulty, Mr. F. W. Harding, a member of the American Society of Mechanical Engineers, devised the support shown in the sketch, which consists of a bracket or stand *A*, which forms the fulcrum; the lever *B* and the carrier *C*. If desirable, the pipe may be suspended by a hanger from the lever instead of the plan followed in Fig. 59.

The effective weight of the lever being equal to the weight of the pipe and fittings, the pipe will at all times be properly supported and free to rise and fall with the expansion and contraction of the boiler.

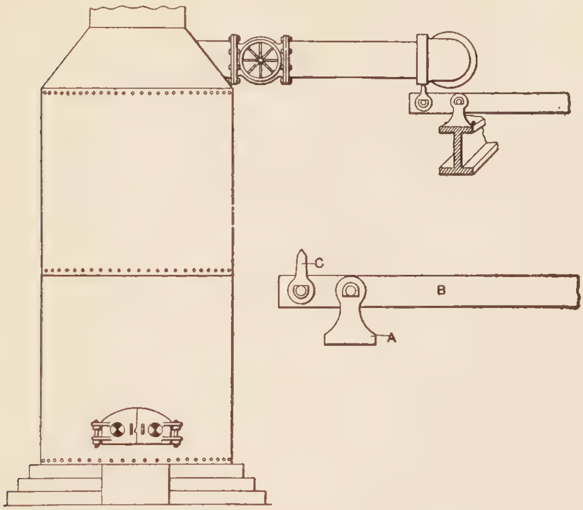


FIG. 59

It will be noted that the flat bearing surfaces of the bracket and carrier allow the trunnions to roll with the least possible friction. In case of difficulty in ascertaining the exact weight of the pipe and fittings, a movable weight may be placed on the lever and adjusted to meet the requirements.

XII

PRACTICAL SUGGESTIONS *

While descriptions of the piping in large plants are always valuable to the progressive engineer, suggestions concerning the location and operation of smaller lines cannot fail to help the working engineer about his every-day duties.

Figure 60 illustrates the 8-inch exhaust-pipe and feed-water heater of a certain steam plant, as originally planned. When erected it was not acceptable for the following reasons. The entire weight rested on cast-iron legs that in turn were placed upon a light floor which it was not practical to strengthen enough to be safe. It was necessary to locate the drip-pipe shown, as high as possible in order to properly drain the exhaust-pipe, as the available "fall" was slight.

The ell was replaced by a tee, as shown in Fig. 61, which rests upon a solid foundation, thus making the cast-iron legs unnecessary, and providing a much better support for the whole, which includes piping above the heater, not shown in the cut. The drip-pipe is now several inches higher than before, although it is still below the bottom of exhaust-pipe. Packing in the three joints between heater and foundation, is now

* Contributed to Power by W. H. Wakeman.

held firmly in place by the weight resting upon it, which is an important advantage, as it is practically impossible to keep the nuts tight, for some of them are not accessible.

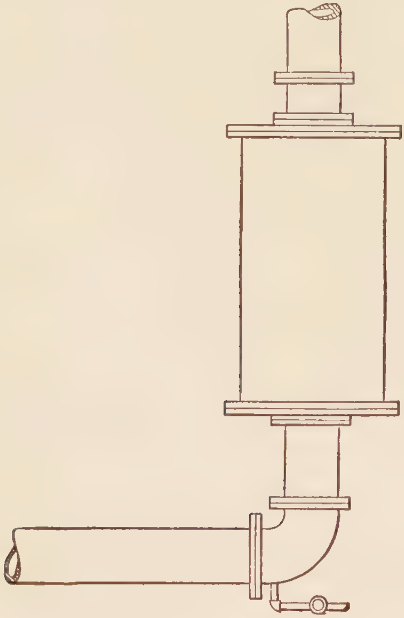


FIG. 60

Figure 62 illustrates another case where heavy piping is supported on a foundation instead of being suspended from floor timbers according to common practice. The lower horizontal pipe, 5 inches in diameter, carries exhaust steam from a heater into the main exhaust-pipe. The larger pipe above it, 8 inches in

diameter, also discharges exhaust steam into the main vertical exhaust-pipe, 10 inches in diameter, which is continued upwards for about 35 feet through the roof.

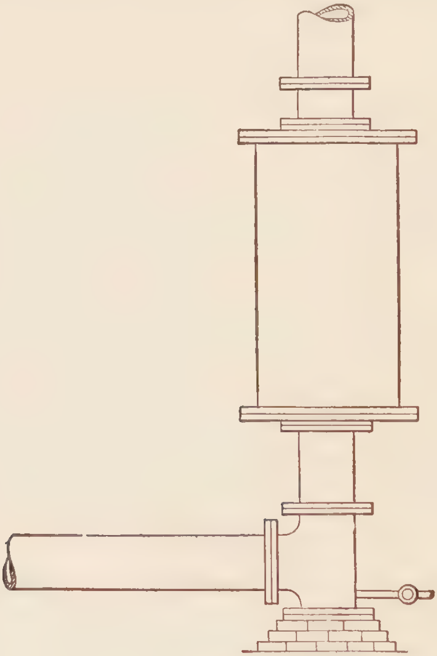


FIG. 61

The smaller vertical pipe shown in the cut is 3 inches in diameter, although a 1-inch pipe would easily have carried off all water coming to it. It was made large in order to serve as a support to the piping above it, and as it rests upon a solid foundation all of the joints

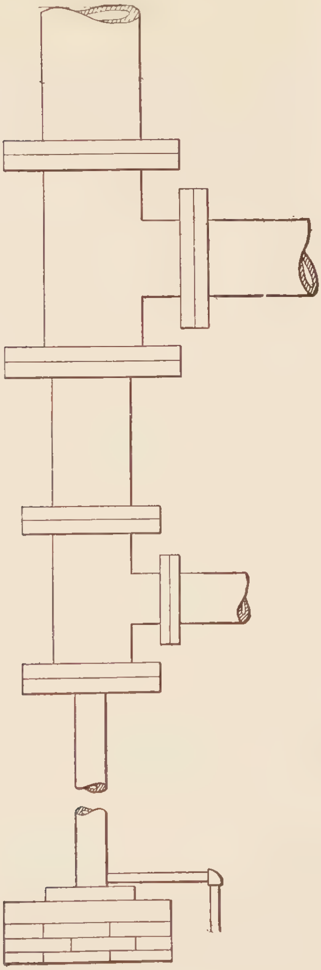


FIG. 62

shown are made secure (against a light pressure) by the weight resting upon them. As the vertical pipe is located in a corner, it will be plain that the bolts in these joints cannot be "followed up" at pleasure.

Figure 63 shows part of a steam drum 10 inches in

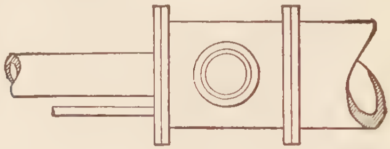


FIG. 63

diameter into which steam from a pair of boilers is discharged through tees, one of which is shown in the cut. Steam is taken out through the 5-inch pipe at the left, and as the bottom of it is much above the bottom of drum, it leaves a "pocket" for the accumulation of water, for the only drip provided is connected into the head of drum as illustrated, and even this is not arranged to act continuously, as it supplies steam to run a pump, and when this is shut off the drum is not drained, therefore is dangerous. A drip-pipe should have been connected into the bottom of drum and attached directly to the boilers so as to provide an automatic drain at all times.

Figure 64 illustrates a 3-inch pipe in which water hammer caused much trouble. The small upper pipe supplied steam for an injector, but the drip-pipe below it was not put in when the line was first used. This proved a partial remedy, but did not eliminate all danger, the cause of which will be understood by

reference to Fig. 65, which is the line connected into the tee in Fig. 64.

This pipe was higher at the right (which is practically a "dead end," as the outlets are closed for a greater portion of the time) than at the drip in Fig. 64, therefore when it was opened, water from this "dead end" flowed towards it, but was met by steam and hurled back with great force.

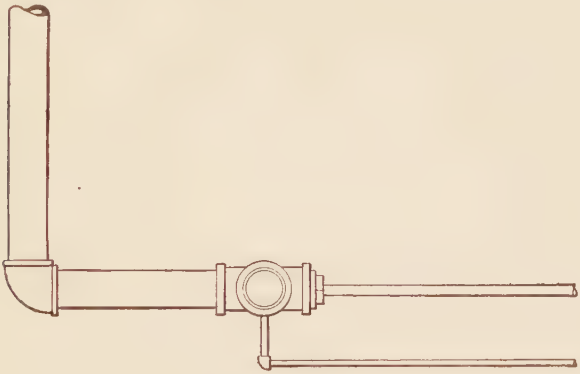


FIG. 64

Putting in the drip-pipe shown at the right in Fig. 65 did not prove a remedy until this end of the line was lowered, the "dead end" becoming the lowest point, when all condensation, even after the drip-pipe had been closed long enough to cool the entire line, could be drawn off without causing water hammer.

It will be noted that the small pipe (shown in Fig. 64) used to supply the injector with steam, is not properly located, as it takes steam from a larger pipe

in which much water collects, and no provision was made for draining it when first put up.

Figure 66 illustrates the blow-off pipe of a steam boiler, as we frequently find them in practice. The first turn outside of the brick wall is an ell, consequently if mud and scale collect in the horizontal pipe which passes through the combustion chamber, there is no way to remove them, except to take down the pipe.

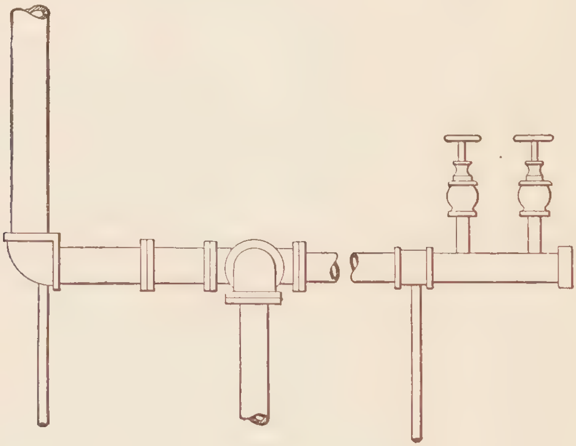


FIG. 65

A gate-valve is located next to the boiler, followed by a nipple and an asbestos packed plug cock. The valve is provided for use in case the cock leaks.

Figure 67 illustrates the blow-off pipe, etc., preferable to the arrangement in Fig. 66 The first turn outside of the brick wall is an ell with a plug in the

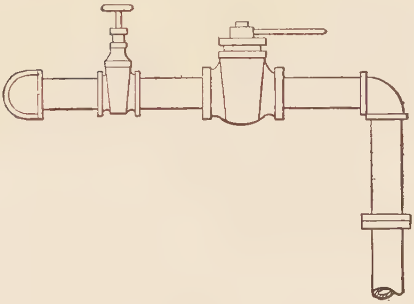


FIG. 66

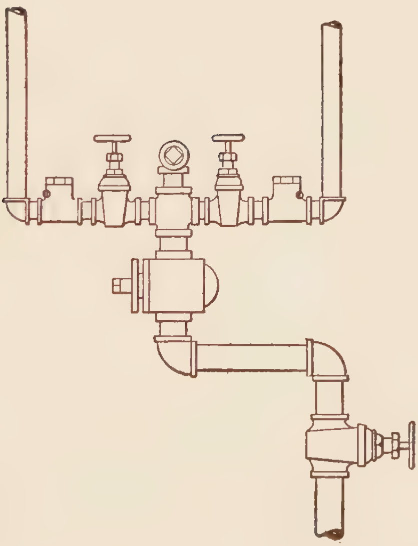


FIG. 67

outer end of it, as shown. When the boilers are washed out, this plug is removed, thus exposing the interior of pipe for cleaning.

The right-hand valves and pipe constitute a drip from the main steam drum, therefore this important part of the steam piping is kept free from water at all times, and this passing in through the blow-off pipe keeps it safe from burning out when the fire is forced.

The left-hand valves and pipe constitute the cold water supply for filling the boiler after it has been cleaned.

The feed-water pipe enters the front head, and discharges at the rear in the usual way. The advantages of this plan are as follows:

Suppose that one of these boilers is emptied, washed out with hot water, and it is necessary to refill it at once. In this case front feed is used, as steam from other boilers heats the feed-water, and the lower half of the boiler is not cooled, consequently no harm results from uneven contraction.

If the boiler is cold, the rear feed is used, bringing in cold water; therefore the lower half of the boiler is not warmed while the top remains cold, therefore no harm is done by uneven expansion. It is just as bad to run hot water into a cold boiler as to run cold water into a hot boiler.

Below these valves is an asbestos packed plug cock, and further down a gate-valve. Have them arranged according to this plan, so that when the valve leaks it can be taken off and repaired, as the cock alone can then be used.

It will be noted that this arrangement of cock and valve is directly the reverse of that illustrated in Fig. 66, the reason for which is that the valve is much more liable to leak than the cock, hence the best one is placed next to the boiler, although some engineers prefer the plan shown in Fig. 66.

When these cocks were first put on the market they were designed so that the plug could be given a complete revolution, but after due consideration of the matter, it was concluded that the best way was to give it one-quarter of a revolution only, and to always turn it through the same quarter, as that is sufficient to open and close it.

This method of operation reduces the leaking to a minimum. This does not agree with the experience of those who turn them any way that they happen to, but this plan seems to give better results than they secure. The improved kind cannot be given more than one-quarter revolution.

Figure 66 does not show the best possible location of the gate-valve, as its stem is in a vertical position. The consequence is that when mud is washed out of the boiler (under no pressure) some of it collects in the bottom of this valve and stays there until the disk descends and packs it more firmly into place. At the same time it prevents the valve from closing tight.

Having tried this plan, found it unsatisfactory and discarded it for that shown in Fig. 67 constitutes a good reason for favoring the latter, in which the stem is in a horizontal position, and no "pocket" is left in the valve to cause trouble.

With either of the above plans in use, when the boiler is blown down under pressure, the gate-valve should be closed first, because, as its disk nears the seat, water rushes through the space at high velocity, thus tending to keep the passage clean.

XIII

STEAM-PIPE CONDUITS

It is not good policy to bury pipe in earth, stone or concrete.

For a temporary job the best plan is a wooden box about 2 inches larger all around than the pipe, with stands made of flat iron to support the pipe in the center of the box, as in the sketch, Fig. 68.

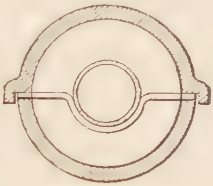


FIG. 68

After the pipe has been tested and made tight, fill the box with mineral wool. If you wish to economize on mineral wool, the corners of the box may be filled out as shown by the dotted lines, but unless you are located where the wool is very expensive, one cost will about balance the other.

Do not use wool made from blast furnace slag, as it usually contains considerable acid, which is severe on

the pipe. Get the kind that is made direct from the rock, and you will have better results.

Do not pack the wool, but be sure that the box is full, especially under the pipe, or the wool will settle down and leave the top of the pipe partially uncovered.

As the wool is practically threads of glass, care must be taken to keep it out of one's eyes or cuts or sores on the hands.

Wool is perferable to the sectional covering for this class of work, for the reason that repairs to the pipe are more easily made, and the expansion and contraction of the pipe will not break it up where the pipe moves through the supporting stands.

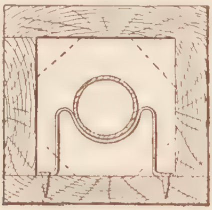


FIG. 69

The cover should be fastened down with screws, so that it may be readily removed without breaking, and brass screws are preferable, as they will not rust rapidly.

Figure 69 illustrates a scheme for underground piping in boiler-rooms, with good success, and there is no reason why it should not be all right for long lines of pipe where a permanent conduit is wanted. It is

not advisible to use the more expensive one of brick or concrete.

It is made up of special vitrified tile, like a sewer pipe split longitudinally, with ball and socket joints, which may be made up with cement for the bottom half. But in order to drain off any water that may find its way into the conduit, it would be advisable to leave a joint free every 8 or 10 feet.

Whichever style of conduit is used, it will be advisable to place a few inches of broken stone under the same to provide for drainage, and provision made for draining away from the trench any water that may collect in it.

XIV

HINTS ON PIPE FITTINGS*

THE engineer often uses up considerable time in hunting around for a right-and-left coupling, nipple, box or flange union in order to complete some connection between lines of pipe that must be joined before the job can be considered complete. A coupling known as the "long screw," shown in Fig. 71, made from a

FIG. 70

piece of pipe and two common pipe couplings, will help him when nothing else is available. The use even of the left-hand pipe die is not required. The connecting nipple is threaded at one end in the usual manner, as at *A*, Fig. 70, and the other end is threaded long enough to allow the coupling *B* and the lock-nut *C* to be screwed away back upon it, as shown. The lock-nut *C* may be made by cutting off a piece of common coupling with the pipe tongs or hack-saw or in a lathe. The connection is made as shown in Fig. 71, where *C* and *D* are the ends to be joined. The end *A* of the

* Contributed to Power by Wm. Kavanagh.

coupling nipple in Fig. 70 is connected to *C* by a common nipple and made up tight. The end of *D* is brought up close to the other end of the coupling nipple, and the coupling *B*, which has been turned on to that nipple far enough to allow the ends to come together, is backed off until it is tight upon *D*. The lock-nut *C* is then turned up tightly against *B* with a piece of lampwick saturated with oil and plumbago between.

FIG. 71

The objection generally raised against the long screw is that it is difficult to keep steam tight, on account of the packed joint and because the threads cut on the long screw are parallel and devoid of the taper which is so essential in steam-tight work. Nevertheless, such a coupling or union has its advantages, because it can be "home made," can be inserted in almost any position or line of pipe, and, when properly put together, will remain steam tight under varying conditions, and gives a finished appearance to the job.

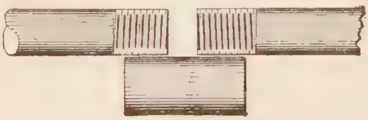


FIG. 72

Probably there is no method employed to connect a line of pipe more substantial and reliable than the right-and-left coupling shown in Fig. 72. In order to

connect up a right-and-left coupling we must first discover the difference between the number of threads the coupling will absorb between the right and left sides; and in order to find this difference we proceed as follows: First screw the coupling on the right-hand thread as far as it is apt to go and be steam tight. When this position has been reached, draw a line on the coupling and pipe, then screw the coupling back and note the number of times that the line on the coupling passes the line on the pipe. Suppose the coupling revolves seven times; now screw the coupling on to the left-hand side as far as necessary to be steam tight, proceed as before, and note the number of turns. Suppose this number to be five. Now subtracting five from seven gives us a difference of two turns, and we see that our coupling must be screwed on to the right-hand thread two turns before being started on to the left. After we have screwed our coupling on the right thread two turns we must then bring our pipe in line and enter the left thread, then screw up with a Stillson or pipe wrench until the coupling is tight. In general, when steam fitters employ the Stillson wrench or pipe tongs to force the difference between the threads that the coupling will absorb, they allude to "expanding the coupling." Generally a good job can be done by finding the difference by screwing the coupling on by hand and noting the difference as before described. Should either end of the coupling leak after it is screwed up, it probably can be made tight by screwing up a little more; if not, the coupling must be unscrewed and another turn or half a turn added to the end that

leaks. The number of threads appearing at both ends of the coupling may not be equal, but this will be of no importance providing the coupling is steam tight. Red lead should never be used in making steam-tight

FIG. 73

joints. Oil and graphite will be found far more satisfactory, and should there be any occasion to break a joint, it can be done with far greater ease and less liability of crushing the pipe.

Figure 73 shows a case where the right-and-left nipple can be employed. After the elbows are in position and pipes *D* and *F* are tried for parallelism, we can then determine the length of our right and left nipple. Of course this length depends to a great extent

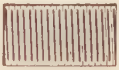


FIG. 74

upon the amount of spring between pipes *D* and *F*. To make such a connection the elbows are threaded one right- and the other left-handed, and the directions given for joining the right-and-left coupling are equally applicable in this case.

Figure 74 is known as the parallel nipple, so called

because there is no taper to the threads. Such a nipple is generally used in making a very close connection between a tee and an elbow, or between two elbows or two tees. It can be "home made" by running the die along a piece of pipe a little over the required distance and then cutting off the right length. This nipple should be used sparingly, as it is very apt to leak and almost impossible to caulk, owing to its position.

FIG. 75

Figure 75 is known as the close shoulder nipple, and in general it can be used wherever the parallel nipple is employed. The use of the close shoulder nipple instead of the parallel nipple insures a safer and more reliable job, as it is free from the defects of the parallel nipple, and it is much stronger because so much of the metal is not cut away in its manufacture.

XV

SIZES OF PIPE

It is strange what a number of good mechanics are to be found who are totally in the dark regarding a subject so closely allied to their own line of work as the sizes of wrought-iron or steel pipe, and the relative thickness of ordinary pipe, extra strong and double extra strong. This fact is illustrated by the following incident. A note was sent to the man in charge of certain work, asking if there was a sufficient supply of double extra strong pipe on hand. The answer came back that there was plenty of all the sizes that were to be used. When the time came to put things together it turned out that there was plenty of common pipe, but nothing heavier than extra strong, and not enough of that. Men often wonder whether the size of a pipe is measured on the inside or the outside, while the fact is that it is neither the one nor the other. The size given means nothing as an exact measurement, and is, in fact, not much more than a name by which it may be known. A $1\frac{1}{2}$-inch ordinary pipe measures about $1\frac{5}{8}$ inches diameter and $1\frac{9}{10}$ inches outside, and is often mistakenly called 2-inch pipe by those who are entirely unfamiliar with such matters. A 2-inch pipe is $2\frac{1}{16}$ inches inside and $2\frac{3}{8}$ inches outside, while

a $3\frac{1}{2}$-inch pipe is exactly 4 inches outside. Every good mechanic should inform himself upon such a subject as this, whether it relates to his own branch of work or not. The possession of such knowledge may prove of value to him, and it costs nothing to keep. Tables giving all the particulars regarding pipe sizes are given in many pocket-books and in numerous trade catalogs. An inspection of such a table will show that the nominal diameter of pipe is an approximation to the actual internal diameter, and that the thickness of extra strong is about one and one-half times that of ordinary pipe, while the thickness of double extra strong is about twice that of extra strong. These expressions are often abbreviated to E. S. and D. E. S., or they are sometimes written X and XX. The outside diameter is always the same, no matter what the weight of the pipe may be; the inside becoming smaller as the thickness is increased. This of course is essential in order that the same taps and dies may be used for all pipe of a given size regardless of the weight. The internal area of double extra strong is, for some of the sizes, not much more than one-half that of ordinary pipe, and this is a fact which must not be lost sight of when considering their relative discharging capacities. For sizes below $1\frac{1}{2}$ inches, the area is much less than one-half.

The larger sizes of pipe, from 14-inch up to 30-inch, are made of even dimensions outside diameter, and are known as O. D. pipe in distinction to those above described, which are sometimes called I. D., or inside diameter pipe, although, as previously observed, they

are none of them made to exact inches or binary fractions of inches. The O. D. pipe can be had in any thickness from $\frac{1}{4}$ inch up to $\frac{3}{4}$ inch, varying by $\frac{1}{16}$ inch, while the I. D. can only be found in the peculiar thicknesses given in the tables for standard, extra strong and double extra strong, unless made to order in large quantities and at special prices. There is also light-weight pipe made to exact outside diameter for low-pressure service, in sizes from 3 inches upwards. This light pipe is usually rolled or peened into the flanges, or is riveted to them.

XVI

HOW TO DISTINGUISH STEEL FROM IRON PIPE

Few users of pipe are able to determine, from its appearance, whether it be iron or steel — in fact, many times when customers have thought they were using iron pipe and, in consequence, declined to accept steel, we have proved to them by tests that they were being deceived. In view of this condition, and the further fact that the bulk of the pipe manufactured to-day is steel, and that it is no longer an experiment, but has come to stay, a brief explanation of how it may be distinguished from iron pipe undoubtedly will be interesting to many of our readers.

In the first place, iron pipe is rough in appearance and the scale on it is heavy, whereas on steel the scale is very light and has the appearance of small blisters or bubbles, underneath which the surface is smooth and somewhat white. When flattened, steel pipe seldom breaks; but if a fracture does occur, it will be noticed that the grain is very fine. Iron pipe, when subjected to this test, breaks readily and shows a coarse fracture, due to the long fiber of this material.

The impression often prevails that steel pipe is exceedingly hard, for which reason they imagine that it is threaded with difficulty and that the threads are

easily broken off. This belief is entirely erroneous, the truth being that steel pipe is soft and tough. Threads on this pipe do not break; they tear off, to avoid which it is necessary that the cutting die shall be sharp and thus cut above the center. Dies suitable for steel pipe can also be used on iron pipe; but blunt dies that will work successfully on iron pipe will tear the threads on steel pipe, owing to the softness of the metal.

XVII

A COLOR SCHEME FOR PIPE LINES

The multiplicity of pipe lines in the modern power plant is confusing, to say the least. Some simple method of easy and certain identification, universally adopted, would be a welcome step in advance. Not only would it facilitate the regular work of the attendants in charge, but it would reduce the probability of mistakes in handling valves, and in times of emergency might prevent serious accidents. Furthermore, when a change of engineers is made, the new man would grasp the situation more quickly, and there need be no interruption of the service, nor even a drop in the efficiency. Such a system would also be of decided advantage to inspectors when making their regular visits — whether for the municipal, insurance, or other authorities.

Some attempts in this direction have been made by attaching labels or tags to valves. The United States Government requires all pipe lines in distilleries to be painted in colors, in accordance with an established system. Something has been done also in power plants in this direction, but so far as the writer knows, no complete scheme has as yet been worked out, or proposed, for general adoption.

The writer was confronted with this problem recently when designing the power and service plant of the new Hamburger department store, at Los Angeles, Cal., of some 1600 horse-power capacity. Here there were not only the usual steam, exhaust and feed lines, but a sprinkler system, iced-water distribution, air lines — both compressed and vacuum — ammonia and brine lines for refrigeration and oil, both as boiler fuel and for lubrication. The solution finally worked out was as follows, previous color schemes being adopted as far as possible:

STEAM

High- and medium-pressure.............................White
Low-pressure heating lines....................Aluminum bronze
Exhaust lines..Gray

HOT WATER

Returns from heating system..................Aluminum bronze
House supply...Maroon
Boiler feed.......................................Bright red
Pure drains from high-pressure and exhaust-head drips.......Pink
Impure drips, overflows and boiler blow-offs, to blow-off tank..Black

COLD WATER

From city mains or deep-well and general house distribution
Light blue
Sprinkler lines including tank, excess-pressure and draining systems...Blue

ICED WATER

Drinking-water lines.......................Dark or navy blue

AIR

Vacuum-heating and house-cleaning lines.............Light green
Compressed.......................................Dark green

REFRIGERATING

Ammonia, Gas	Yellow
Ammonia, Liquid	Bronze
Brine	Orange

OIL

Lubricating system	Light brown
Boiler supply	Dark brown

These colors are to be applied to the pipe lines after completion and test. They will be applied directly to the pipes themselves where they are left bare, and on top of the finished covering for all others.

The pneumatic-tube cash system, being of polished brass pipe, was not thought to need special coloring.

Pipe lines for hydraulic elevators, when installed, might be violet. Still further differentiation, if desired, could be secured by painting the valves and fittings a different color from the pipe itself.

Gas pipes, where exposed, might be left black, as there would be no danger of confusing them with impure drains.

Care must of course be taken to secure colors that will not fade under heat.

The above plan is believed to be consistent and reasonably complete, and is recommended for general adoption. — WILLIAM H. BRYAN, in *Steam*.

XVIII

EFFECT OF SUPERHEATED STEAM ON CAST-IRON VALVES AND FITTINGS *

THE effect of superheated steam of high temperature upon cast-iron valves and fittings is a question that has not been clearly and satisfactorily demonstrated. Nothing definite can be found in the literature on the subject, beyond a few statements which are unaccompanied by any convincing proof.

That high heat materially changes the physical properties of cast iron has been well known for a great many years, such as in the case of gas-retorts, grate-bars, etc., but these are instances where the temperature is much higher than in the case of superheated steam; and whether superheated steam would have a detrimental effect on cast iron at its comparatively low temperature has not been clearly demonstrated. A careful search of all available literature has not been productive of much information.

AN IMPORTANT POINT

As cast iron has been, and is now, more generally used in the manufacture of valves and fittings than any other material, it is imperative to know whether

* From Valve World.

it may be counted on to retain its strength if used in superheated steam lines. It is also exceedingly important, in view of the fact that there is so much of this material now in use in superheated steam work, to know if these goods are likely to give out in a comparatively short time. It is surely a very important question, and one which should be noticed before serious trouble is experienced.

Information on this subject does not appear to be entirely convincing, because it has not been followed up with convincing proof. That is to say, while some observers appear to have discovered that cast iron, after being subjected to high temperature due to superheated steam, for some time, was comparatively weak, there is no evidence showing that they knew the strength of this particular iron before it was subjected to the superheated steam, which is quite a defect in the evidence, from the fact that all users of cast iron know that there is an enormous variation in its strength.

Evidence not Conclusive

So we claim that we are justified in taking the position that the evidence above referred to is not at all conclusive.

Crane Company has been very anxious to obtain something definite on this important subject, and the opportunity for investigation presented itself a few months ago, when a 14-inch high-pressure gate-valve, which had been in service for four years, was taken out and replaced by a steel valve of similar design.

The company, having a complete record of the tests of the iron which was used at the time this valve was made, was in a position to determine accurately what the effect of long-continued superheated steam really was. The salient points are:

A loss of strength in the body of 49 per cent., while the metal in the flanges, not being under quite so high a heat, and not directly exposed to the action of the steam, showed a loss of only 33⅓ per cent.

Following is the detailed result:

Test on 14-inch No. 9 E cast-iron gate-valve, extra heavy, removed from a superheated steam line.

Steam on line, about August, 1903.

Valve taken out, September, 1907.

Time in service, 4 years.

Pressure of steam, 200 lbs. per sq. inch.

Temperature of steam, about 590 degrees, sometimes a little higher.

Original strength of cast iron, as shown by test bars, 22,400 lbs. T. S.

Strength of bars cut from body of valve, 12,303 lbs. T. S., 11,608 lbs. T. S., 10,610 lbs. T. S., 12,440 lbs. T. S.; average strength, 11,740 lbs. T. S.

Loss of Strength Shown

Loss of strength in body after four years' service as compared with original test bars 49 per cent.

Strength of bars cut from the flanges on the valve, 14,900 lbs. T. S., 15,250 lbs. T. S.; average, 15,075 lbs. T. S.

Loss of strength, 33⅓ per cent.

It will be noted that in order to avoid mistakes we made several tests as it will be seen that while the metal in the body shows a loss of 49 per cent. as compared with the original test bars, the actual loss is somewhat less than this, for the reason that the body metal was thicker than the bars, and the cooling strains in a large casting tend to lower the tensile strength of the metal about 10 per cent.

Crane Company has made determinations along this line, and finds, with test bars 1 inch square, showing a tensile strength of 22,000 lbs., it is safe to assume that bars cut from a heavy casting will show about 20,000 lbs. However, taking this low figure, the loss in the body was $41\frac{3}{10}$ per cent., which is sufficiently serious to condemn cast iron for valves and fittings on superheated steam lines having a total temperature of 590 degrees.

The information given here we feel is entirely reliable. We cannot conceive where there can be any error, and if it be true, we cannot see any other outcome in this matter than that a lot of plants that are fitted with cast-iron work throughout the country are liable to get into serious trouble, and it appears to us that all engineers in charge of this kind of work owe it to the companies they are employed by, to look into this matter without delay, and see if our experience is confirmed, which they can do by simply taking out a fitting to see what its strength is.

Of course they may have no data to determine what the loss has been, because the manufacturer probably hasn't any record of what the material was that entered

into it; yet at the same time they can get some idea as to whether the material has deteriorated.

First. The valve was originally $22\frac{1}{2}$ inches long, but when taken out it measured $22\frac{13}{16}$ inches, showing a permanent elongation of $\frac{5}{16}$ inch. The fact that iron, subjected to high heat for long periods, will take a permanent expansion set is well known, but it was news to us and probably to nearly all engineers that this took place at a temperature as low as 600 degrees.

NOTE. — *Numerous tables of Sizes and Weights of pipe are published by manufacturers; and the Tables of Dimensions and Weights of Pipe and Fittings republished from Power by the Hill Publishing Company will be a valuable book of reference to be used with this volume.*

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PUMPS

INTRODUCTION

THE solution for many of the puzzling troubles with pumps which every engineer is liable to encounter will be found in this work. There are also a number of unusual instances of repairs, which may prove of much value where pumps are seemingly broken down beyond use, and these special instances will often suggest other possibilities of the same sort.

Two chapters are given to the important matter of setting the valves of duplex pumps, and much attention is given to this same subject throughout the book. It is believed that a careful perusal of these pages will give the reader a sufficient number of illustrations in the operation and repairs of pumps to handle any difficulty that may arise in the ordinary operation of a steam-pump plant.

The compiler of this volume is indebted to those contributors of Power whose names appear in connection with the various articles, and to the following special contributors whose suggestions and works have been found useful: Earl F. Webster, W. A. Dow and Samuel S. Murdock.

HUBERT E. COLLINS.

NEW YORK, October, 1908.

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I

PUMP TROUBLES[1]

THE first duty of an engineer in a new plant, before an attempt is made to start the pump, is to get acquainted with the discharge and boiler-feed pipes, making sure that all the valves are open that should be open. Of course this is not necessary taking in charge of a plant where the present engineer is going to leave, as in most instances the old engineer will show the new man all he wants to know about the piping arrangement. Where an engineer has been discharged, however, the engineer who takes his place usually has to start up without this friendly aid.

At one small steam-plant, an attempt was made after having the fires started under the boilers, to start the boiler feed-pump, but before doing this what was supposed were the valves on the discharge and feed-pipes were opened in the belief that the arrangement was as is shown in Fig. 1. When the pump was started it was not feeding water to the boilers. Nothing was found wrong. Then the discharge and feed-pipes were inspected and it was found that the valve *A* had been opened the night before. Upon inquiry the information was elicited that the pipe was supplying

[1] Contributed to Power by H. Jahnke.

hot water for the factory and the valve was only opened a few times during the day. The pump received hot water under pressure from a heater, and when hot water was wanted in the factory the feed-valves *B* and *C* were closed and the valve *A* opened for a short time; then the valve *A* was closed and *B* and *C* opened. This appeared to be a bad arrangement, so it was changed by connecting the pipe for the factory directly to the heater, which was of the closed type and received water from the city main.

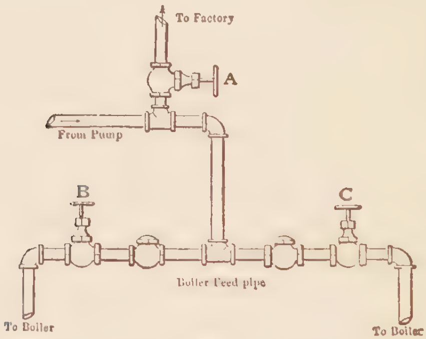


FIG. 1

In some new plants after the discharge and feed-pipes are put in and in use, it is found that the arrangement is not what it should be, and careless engineers, instead of making the needed changes at once, although there has been ample time to think it all out, wait until something goes wrong and then make the changes in a hurry.

Probable Causes of Pump Troubles

If upon starting a pump it is found that it pounds, this may be due to various causes, such as insufficient water supply due to an obstruction in the suction pipe, a leaky foot-valve, a loose water-piston or loose nuts. Also the water-piston may have a tighter fit at some point of the stroke, due to unequal cylinder wear, in which case reboring the cylinder will be in order. Then, again, if the pump is of the duplex type, the pound may be due to improper setting of the steam-valves.

If a pump fails to draw water this may be due to the following causes: If taking water from a well or other supply, either the foot-valve of the suction pipe may leak; the suction pipe may be too small, there may be some obstruction in the suction pipe or in parts leading to the water cylinder; the lift may be too high; the water valves may leak or break; foreign matter may have lodged beneath the suction or discharge valves; the packing on the water-piston may be worn out and leaking; the seats of the suction and discharge valves may be broken, or the water valves may be prevented from lifting because the springs were screwed down too tightly.

If a pump is supposed to receive water under pressure yet fails to get water, it may be due to some valve in the supply pipe not being opened; or the supply pipe may be clogged up; or there is a loose disk or a break in some valve which prevents the full supply going to the pump; the valves may not be wide open;

the supply pipe may be too small; the supply pipe may also furnish water for some other purpose in the factory and may be too small to supply both pump and factory. If a pump receives water under pressure from the city main, the supply pipe should be run direct from the water meter to the pump, and not used for any other purpose, unless the pipe is large enough to supply enough water for both places.

Engineers are often troubled by a groaning noise in pumps; this groaning may be due to any of the following causes: The cylinder oil used may be too heavy (cases are known where the use of lighter oil has cured the trouble); the piston-ring edges may have become so sharp that they scrape the oil from the cylinder walls; if the water-piston packing is too tight, the excessive friction will cause a groaning.

Graphite Mixture Cures Groaning

Groaning in a cylinder can often be cured by the application of graphite mixed with cylinder oil, forced into the cylinder with a hand pump.

Figure 2 shows a good arrangement to place on pumps for feeding graphite and oil to the pumps once a day, or as often as may be necessary. A 2-inch nipple about 5 inches long is provided, with a 2 x $\frac{1}{4}$-inch reducer on each end; one end is screwed directly into the steam-chest by means of a $\frac{1}{4}$-inch close nipple, and on the other end is a $\frac{1}{4}$-inch close nipple with a $\frac{1}{4}$-inch tee. Then a close nipple and valve are placed in the steam-pipe of the pump above the valve, and a pipe is run from the valve to the feeder, as shown. The reason

for placing the valve above the pump valve is to be able to use the full boiler pressure to force the graphite into the cylinder when the pump is throttled down.

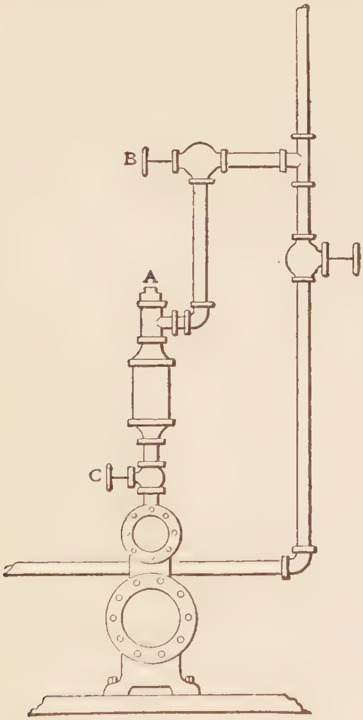


FIG. 2

The operation is as follows: When it is necessary to feed the graphite mixture, the plug *A* is removed from the tee and a supply of graphite and oil placed in the

feeder, the plug is replaced and the valves *B* and *C* are opened, when the graphite will be forced into the steam-chest and cylinder.

Correcting Unequal Stroke

One side of a duplex steam-pump used to make a shorter stroke than the other, due to the water cylinder being worn, and no amount of adjusting of the steam valves would remedy the trouble. Then was tried the following method, which readily overcame the difficulty: The piston and rod were removed from the troublesome side of the pump and the walls of the cylinder were covered with graphite mixed with a little engine-oil, well rubbed in; the piston and rod were replaced and the pump was run slowly with no water in the water end for a short time; then the pump was put in service again, when this side ran much better than the other. The other side was treated in like manner, when both pistons made a full, even stroke, and when the pump was not in service the pistons could be moved easily by hand, which showed that there was not much friction in the water cylinder. It was also found that the water-piston packing will last much longer, and that this treatment is likewise excellent for the steam cylinders of pumps and even engines, and they are so treated whenever the pistons are taken out for any reason.

If a pump begins to run more slowly with the steam valve wide open, if the pump is in good condition, the trouble may be due to an obstruction in the feed-pipes, such as scale.

A short time ago a pump began to slow up every day, with the steam-valve wide open. At first it was thought that the steam-piston was running dry, but feeding more oil did not remedy the trouble; then the check-valves on the feed-pipes were examined, when it was found that the checks were so covered with scale that they could not lift sufficiently to admit the water the pump forced into the pipes, and of course the pumps had to slow up. When the checks were cleaned the pump worked as before. Engineers who use very dirty water for the boilers should examine the check-valves in the feed-pipes whenever the boilers are washed out.

Frequently engineers fail to examine the packing in the water-piston until the pump refuses to supply water to the boilers, or otherwise behaves badly, and when looking for the cause they find that most of the water-piston packing has disappeared and may be lodged beneath the water valves or in the feed-pipes. This is obviously bad practice; the cylinder-head should be removed frequently to see what condition this packing is in. If it begins to show signs of giving out, it should be removed at once and replaced by new packing, no matter if the pump has been doing good service, for if this is not done the pump is liable to fail at any moment.

The water valves should be examined in like manner, and if it is found that they are beginning to leak, valves of hard composition, such as are used in pumps handling hot water, should be refaced at once by rubbing the valve on a sheet of fine sandpaper laid on a

smooth, flat surface. It is also a good plan to have a good set of new valves in stock, as one can never tell when a valve is going to break or meet with some other accident. Another bad practice is to start a steam-pump in the morning, after it has been standing idle for some hours, without opening the drain-cocks on the bottom of the steam cylinder. These cocks should be opened so that the water of condensation will drain out of the cylinder. They should be closed again, of course, as soon as the condensate is blown out of the cylinder.

A Knowles single-cylinder steam-pump, used for boiler feeding, started to give trouble one day by stopping at one end of the stroke. Adjusting the rocker connection bolt so as to equalize the stroke would not help the trouble.

When the pump was taken apart it was found there was too much lost motion between the lug on the slide-valve and where this lug fits into a slot in the steam-chest piston. To take up the lost motion a piece of heavy sheet iron was riveted on each side of the lug as shown in Fig. 3 at *A* and *B*. After this was done and the stroke was adjusted, the pump ran nicely.

In another case a single-cylinder steam-pump received hot water under pressure. One day this pump failed to furnish enough water, and on examination it was found that two of the water valves had broken. When they were replaced by new valves the pump worked a little better, but had to be run at a higher speed than before. Again examining the water end, it was found that the new valves did not seat properly,

due to scale around the valve-studs. Where a pump receives hot water scale will form on the valve-seats and valves and in time will break in some places, when of course the valve will leak and the engineer may look a long time for the cause of the trouble.

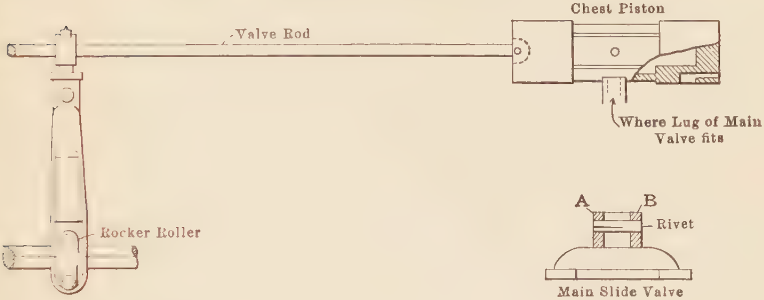


FIG. 3

When putting in new valves, the seats and valve-studs should receive a good cleaning.

It is not advisable to use hot-water valves until they are in bad condition, but they should be taken out occasionally and rubbed over a piece of sandpaper; also overhaul the valve-seats.

A duplex steam-pump which received water under pressure and which is used for boiler feeding, failed to supply enough water. Examination of the water valves and piston showed nothing wrong, so the pump was started again, when it was noticed it was supplying plenty of water. This was a puzzle until it was found that the drain valves from the water cylinder were not closed at night, and when the pump was started

after examination, the drain valves were closed, so that the water which at first went into the sewer afterward went to the boilers.

Some engineers have the bad practice of letting a feed-pump run until it fails to furnish water. Some day they may get caught with low water in the boilers and the pump in bad shape and if there is no other means of feeding, it may be a case of shutting down the plant until the pump is repaired.

II

PUMP TROUBLES[1]

There is probably no machine which is more generally used in plants of all kinds where steam is used for power than the steam-pump, either for feeding boilers or for elevating water for various other purposes, and it is doubtful if there is anything which can cause more trouble and worry to the engineer in charge than a troublesome pump. Still, it would be hard to find a more simple machine, and it is the object of this chapter to touch briefly upon some of its numerous troubles, their causes and remedies.

In the instruction in the manufacturers' catalogues a statement something like this will often be found: "Be sure the water end is all right before disturbing the steam end." They certainly have good cause for inserting such a statement. It is well-nigh impossible for anything to go wrong with the pistons or valves, and yet what more is there in the steam end? Notwithstanding this, men work almost a whole day changing the setting of the steam-valves to make a pump stroke regularly, when the trouble was manifestly in the water end; and after spending several hours in fruitless work, they would have the valves

[1] Contributed to Power by S. L. Brainerd.

set wrong and would not know how to set them properly. If it is a single pump it is beyond the scope of this article to tell him how to set the valves, as every make of single pump requires special directions; but one way, and a good way, to set duplex pump valves is simply this: Remove the steam-chest cover and set both pistons in the middle of their strokes. Then set the valves in the middle of their strokes; *i.e.*, so they just cover both steam ports. Now there is in all duplex pumps a certain amount of lost motion in the valve-gear; that is, the valve-stem travels a certain distance before it can move the valve. Adjust the valve-stems so that this lost motion is equally divided on each side of the valve, being sure the valve is in its middle position. In other words, to set duplex pump valves set everything in its middle position. The valves are properly set now, and unless something slips, you need never trouble about them again.

A pump is harder on packing than an engine, because it takes steam at boiler pressure the full stroke — provided, of course, the throttle is wide open so it can get boiler pressure — while an engine only takes the steam at boiler pressure up to the point of cut-off, after which the pressure and temperature drop rapidly. The writer has charge of six pumps, and one of them, a compound condensing pump, with cylinders and heads steam jacketed, would burn up enough packing on the high-pressure rods to run the whole plant, engines and all, until a little tube was run to each rod and fed cylinder oil on them very slowly. This was a great help. This same pump had a chronic groan in

the low-pressure cylinders, and did some little cutting even with 4 pints of oil in twenty-four hours, until graphite and oil was fed to it, and now it runs six weeks on 4 gallons of oil, without a groan.

It makes no difference whether a pump is of the piston pattern or outside or inside packed plunger pattern; they are all heir to the same troubles, and in a general way the same trouble is remedied in the same way in either style. For instance, if a pump runs smoothly for three strokes of the revolution and jerks back suddenly on the fourth, it always indicates a defective valve in the water end, but there is no way of telling whether it is a discharge valve in the end which the plunger is leaving, or a suction valve in the end which it is approaching, except to remove the hand-hole plates and examine the valves until the one is found which is causing the trouble. Referring to Fig. 4, it is evident that if the suction valve *A* is broken, or if the entire valve-seat is knocked out of the valve deck, as sometimes happens, the water, instead of being forced through the discharge valves *B* against the pressure, will simply surge back through the suction valve *A* and flow into the cylinder through suction valves *C*, thereby relieving the piston of practically all resistance and causing it to jerk back to the head end suddenly. Similarly, if the discharge valve *D* is broken, when the piston moves towards the yoke *Y* it forces the water into the force chamber against the pressure, but when it returns towards the head end *H*, valve *D* fails to hold the water back and it surges around from *B* through *D* into the cylinder,

relieving the piston of resistance, the same as in the case of the broken suction valve.

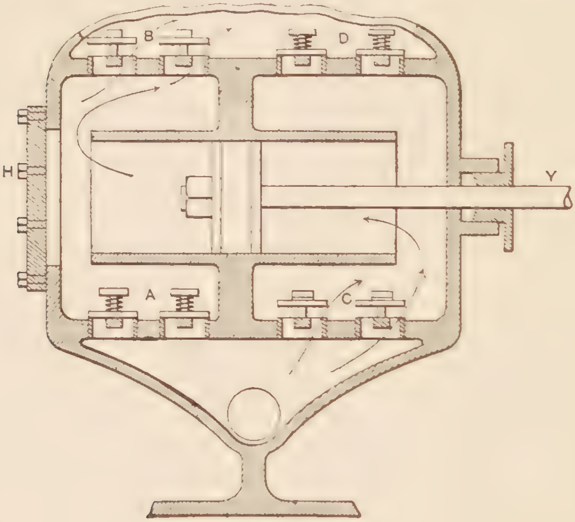


FIG. 4

It is therefore evident that a sudden stroke towards the head end is caused either by a defective suction valve in the head end or by a defective discharge valve in the yoke end, and *vice versa*. A sudden stroke towards the yoke end indicates a defective suction valve in the yoke end or a defective discharge valve in the head end. The degree of suddenness or freedom with which the stroke occurs evidently depends upon the extent of the leak. Frequently a small hole in a valve does not allow enough water to pass to cause

any perceptible irregularity in the action of a comparatively large pump, and it can be detected by the noise of the water surging through it, and should be remedied as quickly as possible. Should a serious defect of this kind exist when the pump is started, it will usually fail to create a vacuum in the suction and to pick up its water.

The maximum theoretical lift of a pump is about 33 feet, but in practice 28 feet is about all that can be relied upon, provided the pump is almost directly over the source of supply. When the suction is carried any distance to the pump, a very good rule for a rough estimate is to consider 100 feet horizontal distance equal to 1 foot vertical lift. However, a 28-foot lift is not to be recommended unless absolutely necessary. It is essential that the suction system be perfectly air tight, especially in case of a considerable lift, in order that a good vacuum may be formed and allow the atmospheric pressure to force the water into the pump. A serious leak may be detected either by the refusal of the pump to pick up the water or by a knock at the beginning of the stroke, caused by the cylinder being only partially fitted with water. Remember that a small quantity of air in the suction expands to a considerable volume when under a vacuum of, say, 15 to 20 inches. When this is drawn into the cylinder and is compressed to 100 pounds or more, it allows the piston to travel some little distance before it strikes the water, thus causing the knock. This knock may be avoided by admitting a little water into the suction through the priming pipe. Whenever possible the

suction pipe should be brought up to the pump at right angles to the direction in which it enters, and then be brought into it with an ell and a short piece of pipe in order to provide a more flexible connection and take up the vibration of the pump. In this way many a leaky suction will be avoided, as the vibration will not act in a direct line with the pipe; and this will more than offset the slight additional friction caused by the turn, especially if a long radius bend is used. The writer knows of one instance where a 12-inch cast-iron suction was cemented rigidly into a heavy wall; and although the pump did not appear to vibrate more than usual, the flange was broken off within a week. A suction with lead-calked joints is particularly troublesome from leaks caused by this vibration.

In order to cushion the pulsations in the discharge an ample air chamber should be provided, and it should by all means be fitted with a gage glass, so that the amount of air in it may be seen at all times. The makers frequently do not provide for this; but if they do not, the engineer will do well to put one on, because these air chambers very often get filled with water and then are of no service. They should at all times be at least half full of air. Very often a heavy water hammer will occur at about mid-stroke on high-duty pumps, if the air chamber is allowed to fill with water, and this is very hard on the pump in general. If no better provision is made for getting air into it, tap the suction at the pump and put in a ¼-inch pet cock, and open this very little until the proper amount of air is obtained, but be careful not to open it too wide, or the

pump will knock from air in the suction. Sometimes it is necessary to run with this pet cock open most of the time.

As a general thing pumps are packed entirely too tight, causing excessive wear on the bushings and plungers and consuming an enormous amount of power. Engineers would do well to give more thought and attention to packing generally. A good way to pack a piston pump is to make a light brass or cast-iron bull-ring to fit loosely over the piston and with just spring enough to hold the packing in position, and cut the packing with lap joints, just like snap rings in a steam-piston, allowing a very slight side play between it and the follower. The water will get under it and set it out, insuring a good joint with loose packing. Packing applied in this way wears longer than when clamped tightly between the followers. Sometimes in packing small pumps the packing is so stiff and hard as to make it difficult to bend it and put it in place. By soaking it in hot water a few minutes it becomes quite pliable. Personally, the writer prefers the outside-packed plunger type to all others, for several reasons. In the first place, the heads do not have to be removed for packing and inspection, and on large pumps this is no small item. If the packing is leaking it is seen at a glance, and the glands can be set up, which would not be the case in an inside-packed pump. The soft packing is much easier on the plungers than the hard packing usually used in other styles, and it may be run just barely tight enough to prevent leakage.

It is a mistake to run a pump without lubricating the water end rods and plungers, as well as the steam end, whenever possible. A great many engineers claim the water lubricates them sufficiently, but after trying them with and without oil, it is found that the packing will last much longer and the rods and plungers will be in much better condition when oil is used, to say nothing of the saving in power. Pipe up the water end for oil the same as the steam end, and frequently dust fine graphite on them with an ordinary squirt-can and the surfaces soon become coated with it, and it does not wash off readily.

III

PUMP TROUBLES

Figure 5 illustrates one side of a duplex pump fitted with a gland for the stuffing-box that is held in place by nuts on two studs. When the packing in such a stuffing-box begins to leak steam, the engineer proceeds

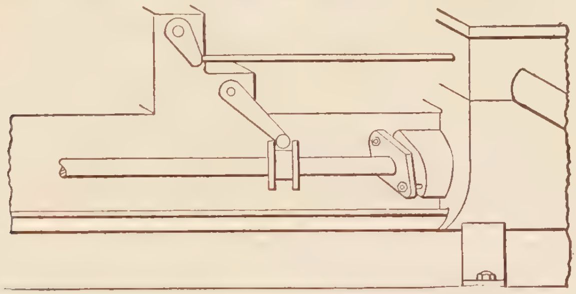


FIG. 5

to screw these nuts on further, but he does not always deem it necessary to turn them both alike, hence the gland binds on the rod, scoring it until it becomes fluted instead of round.

Whenever it is necessary to tighten these nuts, it is a good idea to light a candle and hold it close to the face of gland, then adjust the nuts so that the rod will be exactly central in the gland.

On some of the pumps furnished us these studs are too short, for when packing the stuffing-box the last ring may not go wholly into the box, unless a reasonable pressure is applied to it, and this cannot be done unless the studs are long enough to project through the gland while it is still about ½-inch distant from the box.

Pump manufacturers are beginning to understand this and make the studs accordingly longer, but some of the older ones are defective in this respect. Of course it is possible to get a hardwood stick and a steel hammer and drive the packing into place, but that is an antiquated scheme that should be discarded.

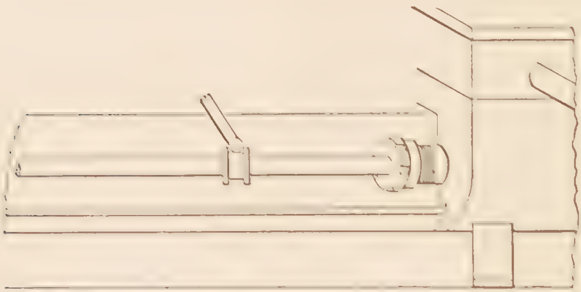


FIG. 6

Figure 6 illustrates another style with which it is impossible to make the above-mentioned mistake, because the gland is much smaller and there are no studs on which to adjust nuts, but instead there is one large nut that is screwed on by means of a spanner wrench.

This nut should be deeper than the length of the gland so that if the packing cannot be pushed entirely into the stuffing-box with the fingers only, the nut will lap over the gland and catch the thread, thus forcing the packing evenly into place.

If the rod is in good order, there is no need of using much force on either kind of gland as a light pressure should answer every purpose. If this does not keep it tight after the packing has been in use for several months, take it out and put in new. It does not pay to use packing until it becomes hard, as it injures the rods more than the value of new packing. When taking out any of the packing, remove all of it, even though it be difficult to get the bottom ring out.

We hear much about pounding in steam-engines, but less about the same trouble with pumps, and the various remedies applied. A pump used for raising cold water, which had previously performed its work very quietly, began to pound in the water cylinder at the end of one stroke, annoying the occupants of a building. Observation taught the engineer that the work of this pump had been increased, and as more water was called for the disagreeable noise grew worse.

He reasoned out the matter as follows. When only a small quantity of water was wanted, the piston did not travel full stroke, therefore the water cylinder was not worn smooth throughout its entire length. When more water was wanted the speed of piston increased and the extra momentum caused a longer stroke; therefore the piston traveled over the comparatively rough part at the end of cylinder, which may have

been a trifle smaller than the middle, causing the piston to bind at this point, hence the pound. Taking the cylinder head off, he proceeded to scrape the end of cylinder smooth, and on starting the pump the pound was no longer heard.

It is a good idea to put an air chamber on the discharge pipe of a pump, especially if the water comes to it under pressure, but such a chamber is often worthless, because there is no glass gage on it to enable the engineer to tell where the water level is; consequently the air space fills with water and it is not observed.

A long glass is not necessary because a short one will answer the purpose if set low, as shown in Fig. 7. If the water level is kept low enough to show in this glass, there will be a good body of air above it to act as a cushion for the pump. When the water level rises too high, draw water out of the air chamber and open the pet cock at the top to allow air to flow in. It will be necessary to draw the water down out of sight, because pressure compresses the air, forcing the water level upward.

A duplex pump was used to deliver cold water against a heavy pressure in a mill. One of the steam-pistons struck its cylinder-head at the end of each inner stroke, causing a heavy pound, the cause of which could not be located until the engineer discovered a break in the cylinder-head gasket, as shown in Fig. 8. Both cylinder-heads were cast in one piece and the packing was also in one piece originally, but a small part had broken out, probably when the double head

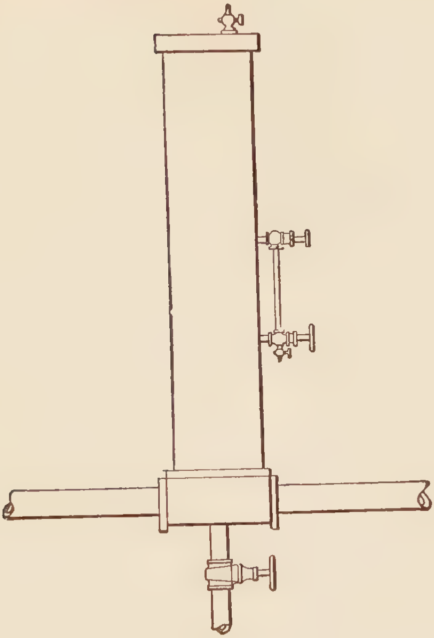


FIG. 7

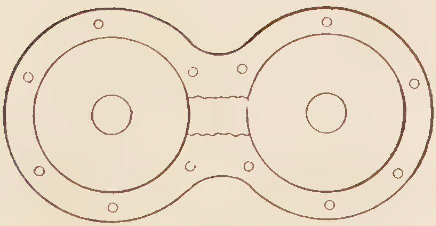


FIG. 8

was removed for inspection, and a new gasket had not been put on, consequently the steam which should have cushioned the piston passed through the small passage to the other cylinder, allowing the piston to strike the head solidly. A new gasket cured the trouble.

A single direct-acting pump began to pound, and the trouble increased as time advanced. The engineer tried to remove the disagreeable noise by adjusting the valve-gear in different ways, without success, after which he had a new valve made with about $\frac{1}{8}$-inch more inside lap, as shown at 2 in Fig. 9, and when it was put in place and steam turned on the pound was no longer heard.

The insurance inspectors came around one day and wished to test the duplex fire pump, and of course they were accommodated. It was run quite fast during the test, taking water from a cistern. After the test was concluded one piston made a stroke much quicker than formerly, denoting a leaky water valve. The water chest of this pump contained forty valves, and under the last one to be examined a stone about the size of a walnut was found. One of the hot-water pumps showed the same defect, and under the first water valve a piece of wood was found. When these were removed the pumps worked perfectly.

A strainer that was packed full of very fine sand caused much trouble. It was at the bottom of a driven well, hence escaped detection for a long time.

In another case the supply of water in an open well was not sufficient; hence the pump sucked air nearly every day, and acted strangely accordingly.

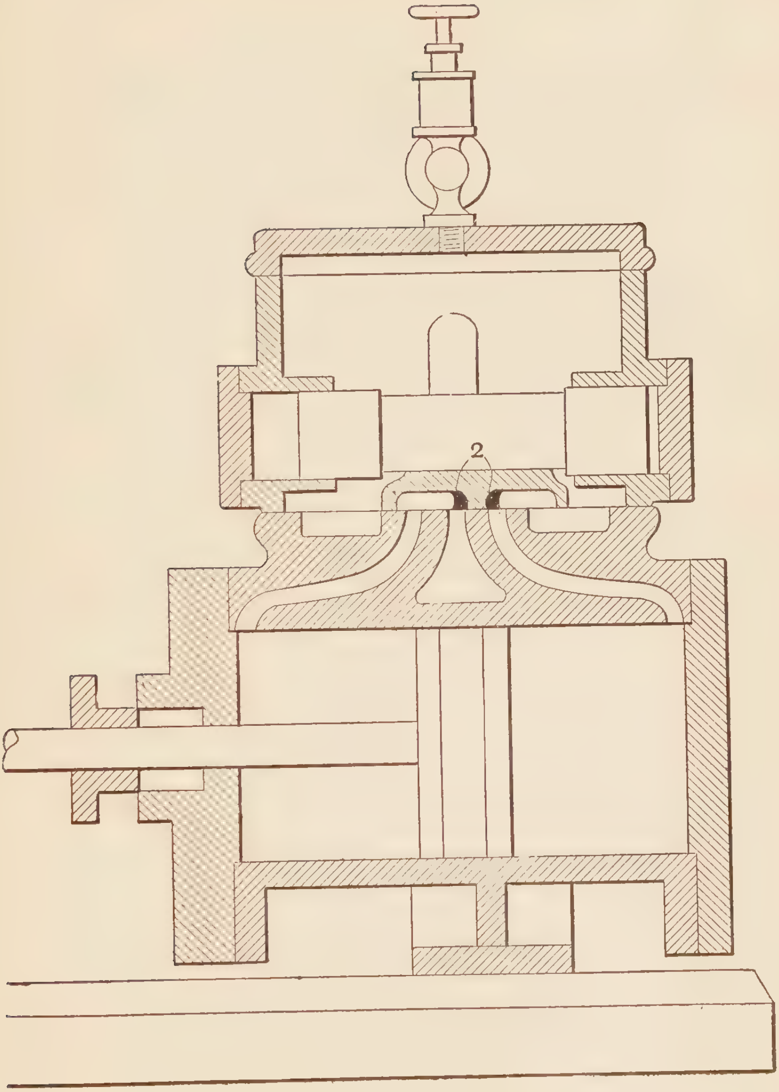


FIG. 9

A crank and fly-wheel pump fitted with brass cylinder, piston, piston-rod and valves complete could not be cured of the habit of groaning caused by pumping very hot and very cold water alternately.

Another pump, used to raise water out of driven wells, could never be made to deliver it against pressure, because it contained a large quantity of air, although no leaks could be found in the suction pipe. When this water was delivered into a cistern, the end of the discharge pipe being below the surface, bubbles of air constantly arose to the surface. After the air had escaped the water was taken out of the cistern and delivered against pressure without trouble.

A pump with no outside valve-gear began to be less and less reliable, and finally refused to work. Careful examination of all the internal parts failed to disclose a defect. As it was a new machine, the manufacturers were notified. They sent a man to remedy the trouble, who brought another steam-chest and auxiliary cylinder with him, which he proceeded to substitute for the defective parts, as it was plain that steam reached both sides of the auxiliary piston, hence it failed to move. He also brought a fine, cup-shaped strainer, which he proceeded to place in the union on the steam pipe.

When steam was again turned on, the pump resumed operations promptly, and has worked well ever since. It is necessary to clean this strainer about once a month, as sediment collects in it, until enough steam to run the pump full speed cannot pass through it. The sediment is sharp and gritty, and is nearly equal

to emery for cutting purposes. The strainer is made cup-shaped in order to provide sufficient opening through it to equal the area of the steam-pipe. There is no mystery about the presence of this sediment while the pipes were new; but why should it continue to collect at this point?

One engineer reports that he removed a similar strainer when he found that sediment collected in it and prevented the free passage of steam; but this was evidently a mistake, because it is much better to clean the strainer once a month than to allow the destructive sediment to pass into the steam-chest and cylinder.

IV

PUMP TROUBLES

A SINGLE-CYLINDER boiler feed-pump after being repaired was run for a few months, when it was noticed that the steam piston would not make a full stroke, but stopped about two inches from the cylinder-head. When the drain-cock was opened the piston would finish the stroke. When the cock was closed, after a few strokes the same old trouble appeared. Everything was in good condition at the water end, and the steam piston did not leak.

When the pump was taken apart everything was found O. K., except at the back end of the steam cylinder there was some gummy substance between the piston and the head. The cylinder oil, which was of a very heavy grade, was responsible for the gummy substance, mixing with the water. This matter could not work out of the cylinder, except by way of the drain-cock. When the cock was closed, of course it kept the piston from finishing the stroke. The piston and cylinder were cleaned, a lighter oil was used and there was no more trouble.

This same pump used to give trouble by slacking up or nearly stopping when within 3 inches of the end of the stroke, at each end. After a thorough in-

spection it was found that the water cylinder was not in line with the steam cylinder, and this caused the piston to bear downward at one end of the stroke and upward at the other end. The bad alignment had been caused by leaving a piece of old gasket on the cylinder face of the water cylinder and the frame when the old one had been replaced with new. The cylinder face was cleaned thoroughly and the parts put together again and the pump ran all right.

Another pump refused to furnish water enough for the boiler. Upon examination, it was seen that the valves at the water end were worn out. New valves were substituted, but the pump did no better then. The water-piston was packed but to no avail. At last it was found that there was scale around the valve-studs and seats, and, the holes in the new valves that had been put in being smaller than those in the old valves, the valves did not properly seat. The scale was removed and plenty of water came after that.

In a good sized station a pump was found with nothing but an ordinary oil cup placed on the steam chest. The pump ran unsteadily and groaned at every stroke. Fig. 10 shows a lubricator as it was fixed up and put on the pump. There is a union at *A*, and another union can be placed between the steam-chest and lubricator, but was not required in this case.

The crack *B* was caused by water freezing in the steam cylinder of the pump. It was patched, as shown in Fig. 11, and then a composition cylinder belonging to the water end of another pump was used and fitted to the inside of the cracked cylinder. The piston was then

turned down and new rings fitted to it. At present the pump is giving good service.

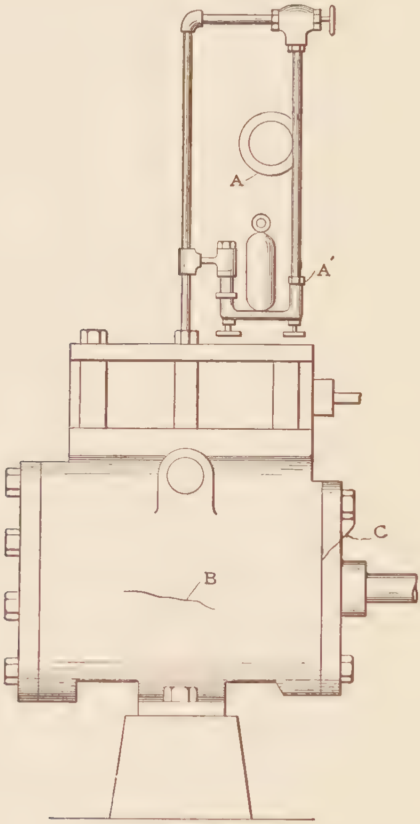


FIG. 10

Once the pump man let the water get below the suction of the pump, with the consequence that the front

head was cracked as at *C*, Fig. 10. A clamp shown in Fig. 12 placed around the cylinder proved effective.

Figure 13 shows an old wrinkle, but it may be new

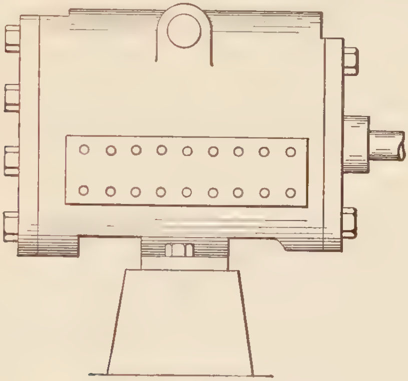


FIG. 11

to some. *A* and *B* are check-valves closed by the atmospheric pressure; *C* is an ordinary globe valve. When starting the pump, air becomes trapped between

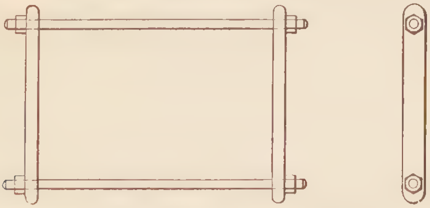


FIG. 12

the delivery valve deck and the suction valves, breaking the vacuum by expanding and contracting in the suction pipe. By opening the valve *C* the air is dis-

charged when starting. As soon as water comes from the valve *C*, close it and the pump will be working properly.

In an isolated plant was found a water supply pump just out of a repair shop giving trouble. On examina-

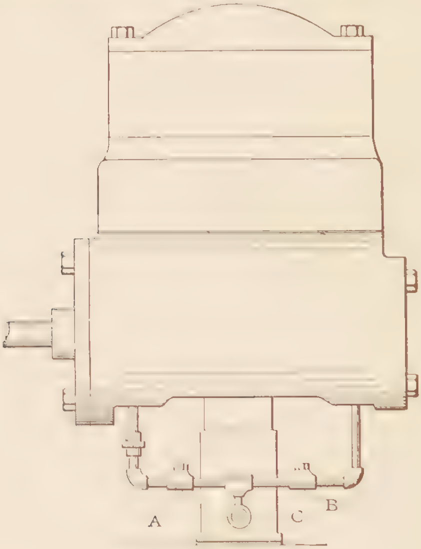


FIG. 13

tion, it was found that the auxiliary valve had not been repaired and was leaking badly (Cameron pump), which prevented its working. The engineer was told what was the matter and that it would be necessary to take the steam-chest back to the shop and bore it and make a new valve. Then as his tank was empty and he needed water if possible to get it, the following plan

was executed. Taking the two heads off the steam-chest, the steam was turned on, noting the amount that leaked through each end; then, after removing

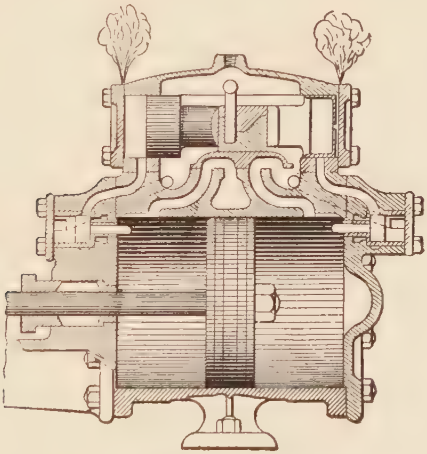


FIG. 14

the gaskets, putting the heads back and screwing the bolts just tight enough to allow the same amount of steam to escape, as in Fig. 14, the pump immediately went to work.

V

SOME PUMP REPAIRS

A CARELESS engineer had started an upright plunger pump in an ice plant with the valves on the delivery line closed, breaking the six-foot cast-iron cylinder into three pieces as shown by the cracks *A–B* and *C–D* in Fig. 15. As the ice plant depended upon this pump for its water, waiting for a new casting from the factory, or the possible longer wait of having a pattern and casting made in one of the local foundries, was out of the question. The warmer city water could have been easily turned in, but every one experienced in ice making and refrigeration knows what a difference of about 20 degrees in the water supply would mean when a plant is pushed constantly to its utmost limit. It would either cut off the greater part of their ice output, or raise the temperature in the cold storage when it was already so high as to be risky. So the engineer was called on to do something.

Two clamps *E* were fitted around the cylinder and four strips *F* fastened the upper pieces to the lower. Then small holes *G* were drilled at intervals all over the cylinder, the outer parts of the holes being countersunk. Then the outside of the cylinder was wrapped with paper and packed around with damp sand in a board casing. A wooden core an inch smaller than

the former bore of the pump was centered on the inside of the cylinder and red-hot babbitt poured in. It was expected that it would be a difficult job to get a perfect cast, but the first attempt proved to be a

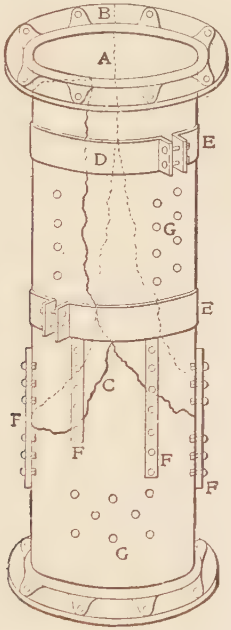


FIG. 15

complete success. It was a simple job to bore this out a little and turn the piston down to fit it. A few burrs of babbitt were cut from the outside and the job was then neat and strong, and the pump gave better service than it had given for years.

Repairing a Pump

Having occasion for the use of another pump, the management decided to buy a new one. Most engineers would have been very willing to have the company go to this expense, but this firm happened to have in their employ one of those careful, saving engineers that are rarely found. This engineer found in the company's pile of junk a pump that had evidently been discarded on account of a crack about 3 inches long in the water cylinder.

Some one had tried calking, but had only opened the crack worse. Somebody, probably the same person, had also made a sort of pocket of clay and attempted the repair by pouring in melted babbitt, but the shrinking of the metal when cooling had been enough to spoil the job.

As the shape of the casting made patching a very difficult undertaking, the engineer decided to try another way. After some difficulty he succeeded in removing the babbitt. He then ground a piece of sheet copper very thin at one end and forced it into the crack. The edge of the copper was then filed off even with the cast iron, and a clamp made of heavy iron was placed around the cylinder and drawn up by the bolt provided for that purpose. The crack was perfectly tight and held for about eight years.

Not long ago this crack again began to leak and no amount of tightening on the clamp would stop it. Another engineer had taken the place of the one who had made the repair. The new man removed the

piece of copper and prepared another like the first had been. But after drawing up with the clamp, the crack still leaked, when pumping against more than 80 pounds pressure. As the crack was nearly in a straight line, the new engineer than decided to try another method. A $\frac{3}{16}$-inch hole was drilled along the crack about $\frac{1}{4}$ of an inch. Then a $\frac{1}{4}$-inch copper wire about $3\frac{1}{4}$ inches long was tapered about a thirty-second of an inch and driven into the hole. When the clamp was again tightened, the crack was water-tight against any pressure, and it looks now as though it would hold as long as the pump would last.

After five years' use a set of 150 valve disks needed facing. They were made of hard composition and had been turned over once, so that they were much worn and had ridges in them from $\frac{1}{64}$ to $\frac{1}{16}$ inch deep. A 16-inch bastard file was first used and it was found after cutting three or four, that the teeth of the file were rapidly wearing off. The file was machine cut, and a hand-cut file was procured, but had the same experience with it. Finally the tool shown in the accompanying sketch was made and found to work extremely well.

A piece of scrap cast iron was put in the shaper. With a screw-cutting tool the grooves shown were cut and filled with emery. Extreme care must be used to keep the edges of the grooves exactly even with the top surface of the block. A hole was drilled in the bottom of the block and screwed in a screw-eye, so that the block could be held in a vise. With this arrangement both sides of a valve could be trued up in one

minute. The tool, Fig. 16, lasted for the truing of both sides of a full set of valves, and is still good for a dozen more sets.

The emery will get filled up with the granulated rubber which is ground off, but the emery can be saved by heating a piece of iron red hot and putting the emery and rubber on it. The rubber will soon burn up and leave the emery as good as ever.

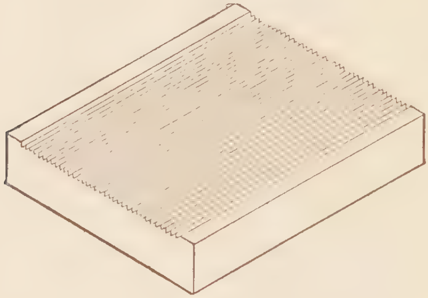


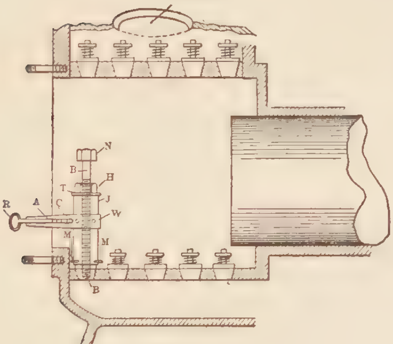
FIG. 16

A duplex pump failed to force water and an examination showed that the valve-seats were badly scored. These seats were expanded into the pump and beaded over so they could not be removed, and to grind them in place was the problem we had to solve.

A 1½-inch coupling was taken, as shown at *J*, Fig. 17, and filled with babbitt and then drilled out for a ⅝-inch bolt *B*. This bolt was threaded nearly the full length. Four small bolts were driven into holes drilled in the lower end of the coupling as shown at *M*. A bar of iron ⅜ x 1 inch was shaped as shown in

Fig. 18, bent as in Fig. 19 and placed at *W*, Fig. 17. The nut, Fig. 20, was made from $\frac{7}{8}$-inch iron and shaped to enter the holes *N N* drilled in the bar, Fig. 18. This is shown in place in Fig. 19 and at *C* in Fig. 17. The nut was drilled and tapped out for a $\frac{5}{8}$-inch bolt *Y*, Fig. 19, the outer end fastened to form a handle as

FIG. 20



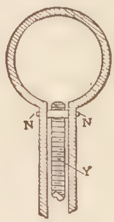


FIG. 19

FIG. 17

FIG. 18

at *R*, Fig. 17. Small holes $\frac{3}{8}$ of an inch in diameter were drilled in the side, forming a circle around the coupling. They were spaced $\frac{3}{4}$ inch apart and are shown in the collar *W*, Fig. 17. An old metal valve was taken and ground good and level on the face. Cavities were sawed and chipped out to allow the four

pins at *M*, Fig. 17, to drop into them, and by making this connection the body *J* could not turn without turning the valve. The lower end of the bolt *B* was screwed into the hole at the center of the valve-seat where the stud holds the valve in place under working conditions. A good coat of lard oil and ground emery was placed between the valve and the seat to be ground, and the lower end of the body *J* was set up snug enough against the valve to allow the coupling to turn and bring a force on the top of the valve. The strap *W* was turned until the taper end of the bolt *C* came in line with a hole in the body *J* at *W*, when the $\frac{5}{8}$-inch bolt was turned in at the handle *R* and this made it fast to the coupling. The bolt *B* was screwed into the seat tightly so that it could not turn. The handle *N* was wound back and ahead, which moved the body *J* and with it the valve. To change the position of the valve, the bolt *C* was backed out and screwed into another hole to put on more energy. The nut *J* was backed off, the coupling raised and the seat examined. It will be seen that by backing out the bolt *C* and putting it into other holes in the body *J* a full revolution can be made. The job was done and well done too, for the old pump throws water good and fast. Fig. 17 shows the tool in place; the valve chamber being an extension of the water cylinder. To grind the suction-valve seats the head was taken off, but the discharge seats were ground through a hand-hole 12 x 7 inches located as shown in Fig. 17. It is well to mention that the strap *W* is free to move up or down or turn around the body *J* when the bolt *C* is not in one of the holes *W*.

Sometimes the thread on the inside of a stuffing-box gland becomes so burred or broken that it can not be started back on the box thread, and it means the taking apart of the pumps to get the necessary repair made unless some method of doing the work in place

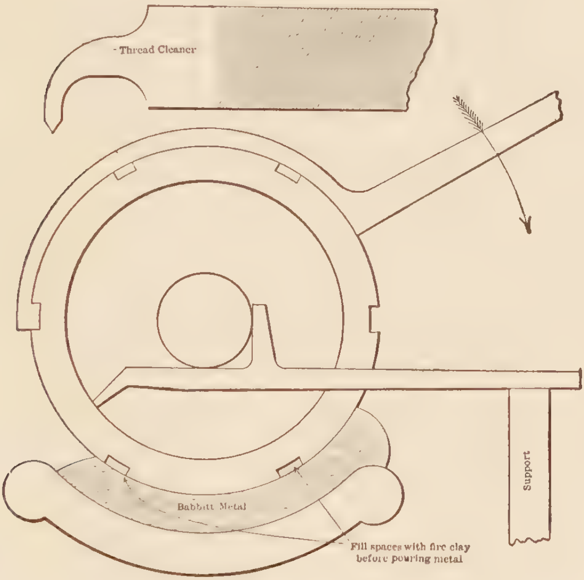


FIG. 21

can be devised. The following method is a good one. Referring to Fig. 21. Move the nut with injured thread out along the rod away from the stuffing-box far enough to get at the thread.

Fill the spaces into which the spanner fits with fire

clay and after packing around the rod to hold the nut central, pour babbitt between the nut and frame as shown in figure. Then make a tool such as shown in cut with a V-shaped point to fit the thread. Then clean out the fire clay from the spanner spaces, and using the spanner to turn the nut apply the tool as shown and clean out the thread. After the thread is clear the babbitt can be cleared out of the way and the nut screwed up in place.

VI

SETTING VALVES OF DUPLEX PUMP[1]

As is well known, the slide valves of a duplex pump have neither outside nor inside lap. This is necessary to prevent the pump from stopping should the valves be in a position to cover all ports. By making the length of the valve the exact distance from the outside edge to the outside edge of the steam port, and the exhaust cavity the exact distance from the inside edge to the inside edge of the exhaust port, there is only one point in the travel of the valve where ports are completely closed; and it is not likely, if it ever should happen that both valves were in this position, that the pump would fail to start off, for the leakage of steam past the edges of the valves will never be exactly the same in all four corners, therefore the equilibrium would be destroyed quickly.

By setting the outside edges of the valves "line on line" with the outside edges of the steam ports, the valves will stand in a central position. If, then, both rocker arms are put in a central or vertical position, the clearance on the valve rod must be the same on both ends. In Fig. 22 this clearance is shown inside of the steam-chest and is marked *C*. On larger pumps

[1] Contributed to Power by F. F. Nickel.

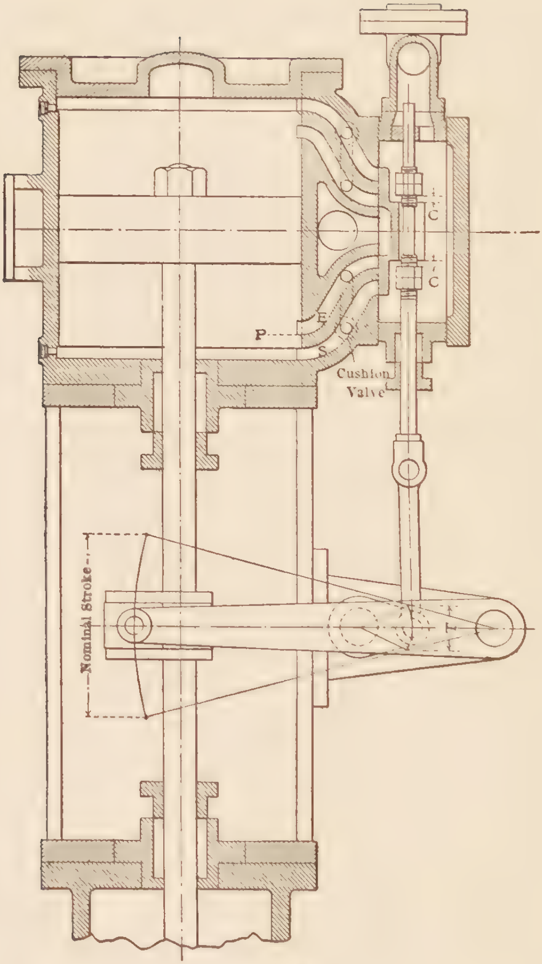


FIG. 22

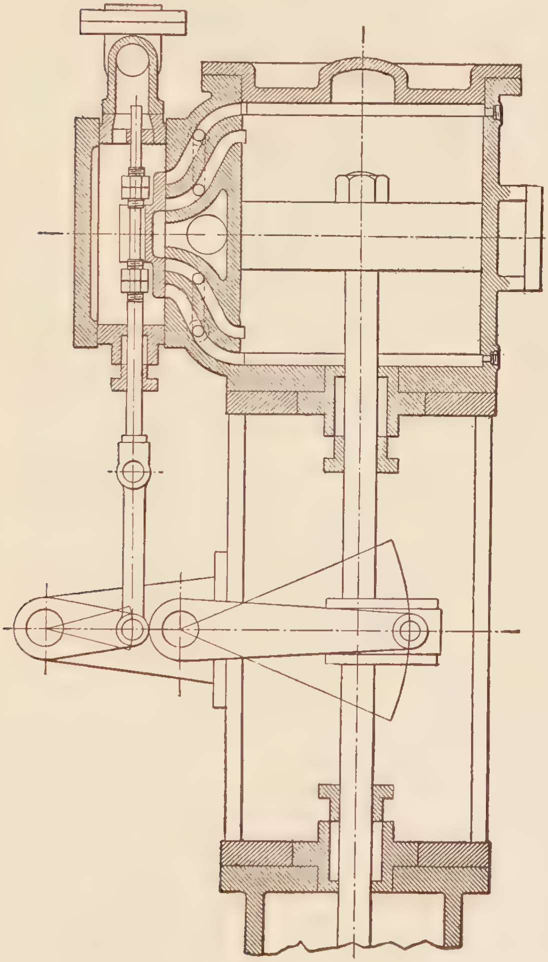


FIG. 23

usually, a lost-motion link is inserted between the crank and the valve-rod clevis, which can be adjusted without taking off the steam-chest cover. No fixed rule can be given for the amount of this clearance, as it must be adjusted to suit the working of the pump.

On a pump of ordinary proportion, such as a boiler feed pump, the total clearance, $2\ C$, should equal about 25 per cent. of the travel T of the crank-pin at nominal stroke. On a low-service pump (also on a pressure pump for moderate pressure) it is often found that the reciprocating parts are so heavy that the cushion, with the cushion valve shut tight, is not sufficient to stop the motion of the piston at the end of the stroke. In this case the lost motion should all be taken up. If the piston does not make a full stroke, the lost motion may be increased somewhat above the figure given, but it must be kept in mind that this will reduce the travel of the valve and the port opening, and thus may affect the speed of the pump.

The Cross-exhaust Valve

In the case of a compound pump there is still another appliance that can be brought into action to regulate the length of the stroke, and that is a connection, provided with a valve, between the two high-pressure exhaust pipes. The object of this connection is to equalize the pressure in these exhaust pipes and make it more uniform. This is called the cross exhaust, and its influence on the distribution of steam is clearly shown by Figs. 24 to 27 inclusive. Figs. 24 to 27, inclusive, are convenient sectional plans of the steam

cylinders of a compound pump, with the pistons in positions that correspond to lines *A B* and *B — C* in the diagram Fig. 29. Fig. 28 represents a diagram with the cross exhaust closed. The steam pressure follows up the full stroke in the high-pressure cylinder,

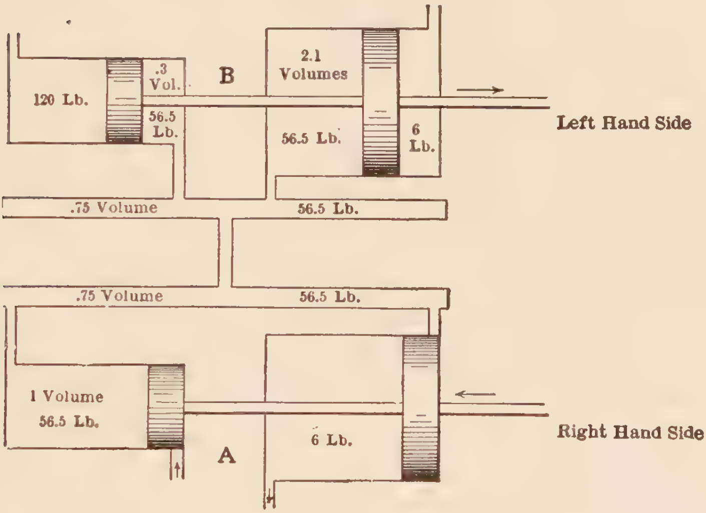


FIG. 24

and when the exhaust valve opens it blows into the intermediate space and mixes with the steam left therein from the preceding stroke.

Assuming the intermediate space to have a volume equal to 0.75 of that of the high-pressure cylinder and a cylinder ratio of 1 to 3, we have the following volumes:

High-pressure cylinder = 1; intermediate space = 0.75; low-pressure cylinder = 3.

Clearances are neglected, as it is only intended to show the action of the cross exhaust. We will also

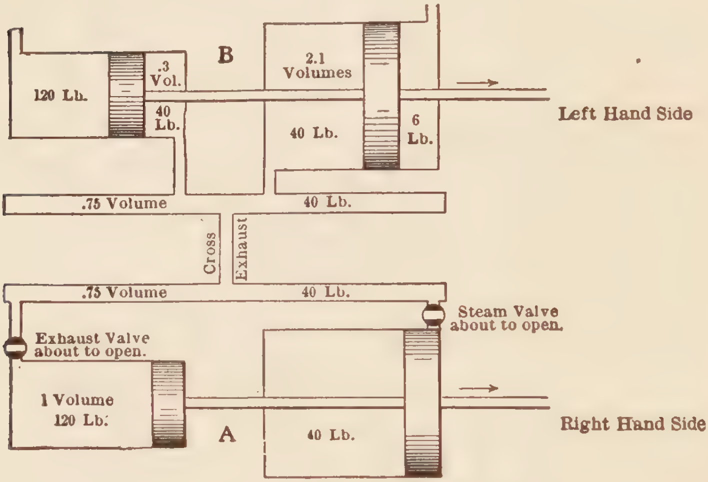


FIG. 25

assume that the steam expands according to Mariotte's law.

$$p \times v = \text{constant},$$

which is sufficiently accurate for our purpose, and assists greatly in getting a clear conception of the behavior of the steam as it passes through the various stages.

The amount of steam passing through one side of the engine is evidently one high-pressure cylinder full

at initial pressure. Its measure is $p \times v = 120 \times 1 =$ 120 lbs. When the high-pressure exhaust valve opens, this steam flows into the intermediate space, where it meets and mixes with steam that was left there from the preceding stroke. This steam was shut off from

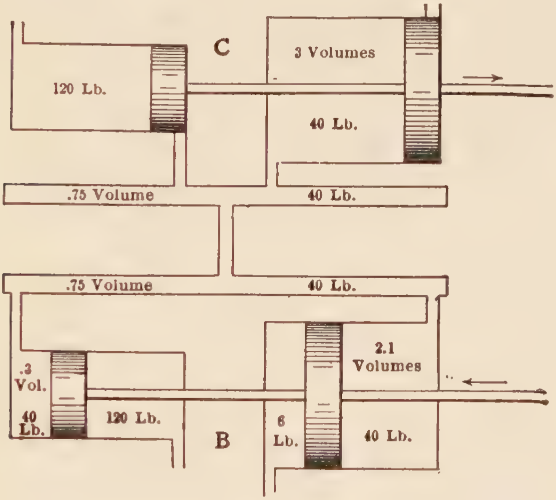


FIG. 26

its communication with the steam in the low-pressure cylinder, when its exhaust valve opened and must be at the same pressure as the steam in the low-pressure cylinder at the point of exhaust. As the ratio of cylinders was assumed to be as 1 to 3, the steam expands three times as it passes from the high-pressure cylinder to the low-pressure cylinder, and the terminal pressure is therefore

$$\frac{120}{3} = 40 \text{ lbs.}$$

It will be noted that 120 is a measure for the steam passing through the engine and this amount is accounted for by the indicator diagram at every point of the stroke. Thus we have:

High-pressure cylinder, $p \times v = 120 \times 1 = 120.$

Low-pressure cylinder, $p \times v = 40 \times 3 = 120.$

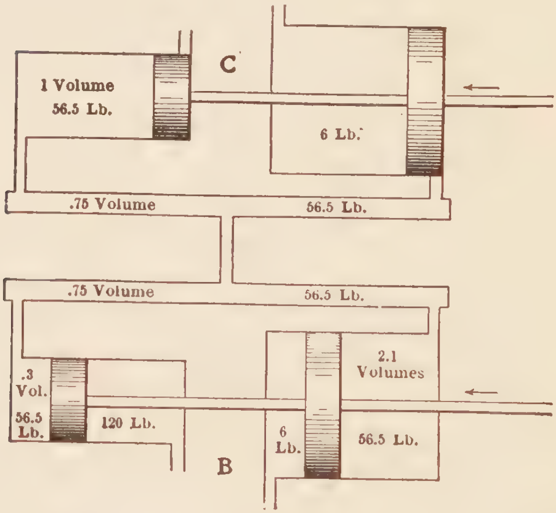


FIG. 27

The amount of steam that is constant and remains in the intermediate space is $0.75 \times 40 = 30$ lbs.; the two combined give $120 + 30 = 150$ lbs., which when distributed over a volume of $1 + 0.75 = 1.75$ results in a pressure of

$$\frac{150}{1.75} = 85 \text{ lbs.}$$

This means that when the high-pressure exhaust valve opens the steam expands from the high-pressure cylinder into the intermediate space from 120 to 85 lbs. without doing any useful work. From 85 lbs. it then expands from the high-pressure cylinder through the intermediate space into the low-pressure cylinder doing useful work upon the low-pressure piston.

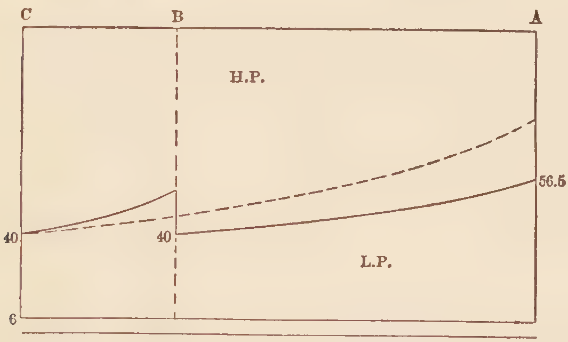


FIG. 28

With two points of the expansion curve, namely, 85 lbs. at the beginning and 40 lbs. at the end of the stroke, it is now easy to construct the remainder of the curve, as it is only necessary to complete the rectangle and draw the diagonal. Where this diagonal meets the line of zero pressure, there is point *o*, the zero point of pressure and volume. Any line drawn through this point *o* will give the volume on the line 85, Fig. 28, and its corresponding pressure on line *A*, Fig. 29.

Under the conditions indicated in Fig. 28, it cannot be expected that an ordinary pump will work satisfactorily, as the following comparison of the steam forces will show.

Beginning of stroke:

$$\begin{array}{lr} \text{H. P., } 120 - 85 = & 35 \\ \text{L. P., } 85 - 6 = 79 \times 3 = & \underline{237} \\ \quad \text{Total steam force} \ldots\ldots & 272 \text{ lbs.} \end{array}$$

End of stroke:

$$\begin{array}{lr} \text{H. P., } 120 - 40 = & 80 \\ \text{L. P., } 40 - 6 = 34 \times 3 = & \underline{102} \\ \quad \text{Total steam force} \ldots\ldots & 182 \text{ lbs.} \end{array}$$

The average of the two, or

$$\frac{272 + 182}{2} = 227 \text{ lbs.},$$

is a measure of the resistance which, in a pump, is constant throughout the stroke. There is, therefore, at the beginning of the stroke, a surplus of 272 − 227 = 45 lbs., and at the end a deficiency of 227 − 182 = 45 lbs. If, however, the cross exhaust is opened, it equalizes these two forces to a certain extent and modifies the diagram, as shown in Fig. 29.

With the assistance of Figs. 24 to 27, inclusive, it is easy to follow the steam through its various stages. In Fig. 24 the pistons of the right-hand side have completed the stroke and are about to return. The cylinders on the other side and intermediate spaces are

filled with steam at the low-pressure terminal, or 40 lbs. The total amount of steam is then

$$\begin{array}{lr} 120 \times \ \ 1 = & 120 \\ 40 \ \times 3.9 = & \underline{156} \\ \quad \text{Total} \ \ldots\ldots\ldots & 276 \end{array}$$

which divided by the volume, 4.9, gives a resulting pressure of

$$\frac{276}{4.9} = 56.5 \text{ lbs.,}$$

as shown in Fig. 25. This increased pressure gives the low-pressure piston of the left-hand side an additional push and enables it to complete its stroke while

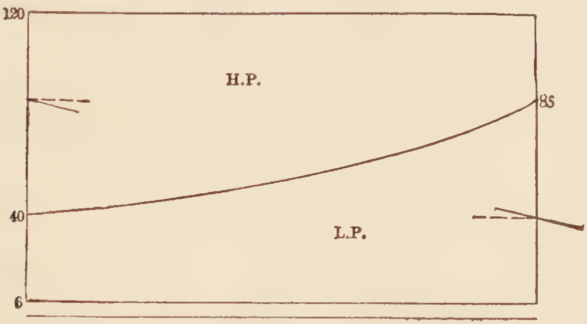


FIG. 29

the steam expands down to 40 lbs. again. Then the steam from the left-hand high-pressure cylinder flows into the intermediate space and raises the pressure to 56.5 lbs., in order to help out the right-hand low-pressure piston.

Fig. 29 shows this action clearly, but in practice

the rise in pressure will not be as abrupt as shown there, as the pulsations in the pipes will still more equalize the differences and produce a practically uniform pressure in the intermediate space.

It will also be noted that by opening the cross exhaust, pressure is removed from the low-pressure piston and shifted over to the high-pressure piston which results in a loss of power and reduced speed of the pump.

The cross exhaust should therefore be kept closed whenever the pump runs fairly well in this condition.

VII

ANOTHER METHOD OF SETTING DUPLEX PUMP VALVES

In setting the valves of a duplex pump, first remove the steam-chest cover; next move the piston-rod toward the steam-cylinder head until the steam-piston strikes the head solid, and make a pencil mark on the rod at the face of the steam stuffing-box, as shown in Fig. 30 at *A*. Now move the piston to the opposite

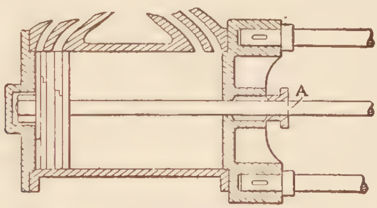


FIG. 30

end until the steam-piston strikes solid, and make another scriber mark exactly half-way between the first mark and the face of the steam stuffing-box, as shown in Fig. 31 at *B*. Then move the piston-rod back until this second mark comes flush with the face of the steam stuffing-box, and now the piston will be at mid-stroke, as shown in Fig. 32. Now take off the steam-chest cover, and place the slide valve exactly

in the center or over the steam ports, and set the slide-valve nut exactly in the center between the lugs on the valve, as shown in Fig. 33. Screw the valve-stem

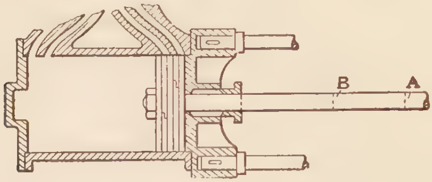


FIG. 31

through the nut until the eye of the knuckle joint is in line with the eye of the link, then slip the link into place. Now we have the valve set on one side and

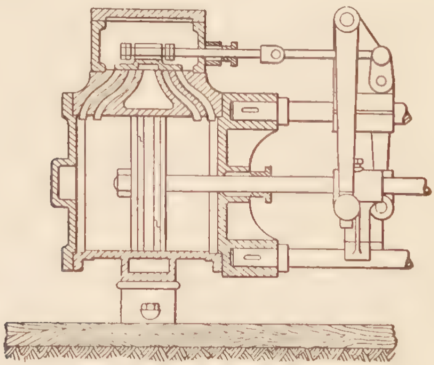


FIG. 32

after repeating this process on the other side of the pump, the valve setting will be completed.

Some pumps are provided with two nuts on the valve-

stem on each side of the valve lugs, instead of one nut between the lugs, as shown in Fig. 34. These are set

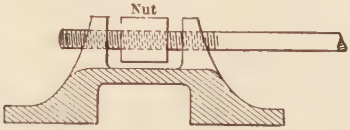


FIG. 33

and locked from the outer faces of the valve lugs, allowing a little lost motion on each side of the lugs. Some make this lost motion equal to half the width of

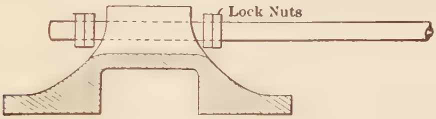


FIG. 34

the steam ports, but sometimes this gives the pump too much or too little stroke, and must be changed accordingly.

VIII

A CENTRIFUGAL PUMP TROUBLE

THE equipment of a centrifugal pumping plant consisted of a 10 x 30 Corliss engine and a 10-inch pump with boilers and accessories, which outfit was to throw 4500 gallons per minute to a hight of 50 feet, 12 feet of which was suction lift.

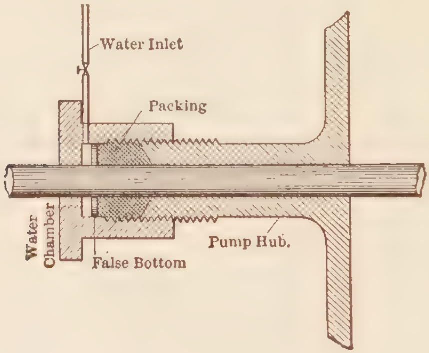


FIG. 35

The pump was speeded so high that the stuffing-box could not be kept tight and cool at the same time. The speed could not be reduced, and the packing burnt out repeatedly. Water and oil applied in the ordinary manner failed to overcome the trouble.

After several days of delay, the gland was taken off and a false bottom inserted on the air side of the packing, leaving a chamber about $\frac{3}{8}$ of an inch deep. This chamber was tapped for a $\frac{1}{4}$-inch pipe, a valve was put on at the gland and the pipe connected into the main discharge of the pump. This practically made a water-packed pump; the packing was left loose and the pump forced to take water instead of air. We had no more trouble with the gland and the packing lasted almost indefinitely.

The change is illustrated in Fig. 35.

IX

BOILER FEED-PUMPS[1]

In selecting a feed-pump two factors enter into consideration, namely, capacity and speed. By capacity is meant the average quantity of water that the boiler which the feed-pump is to supply is capable of evaporating in a certain time, and it is clear that the feed-pump selected should be large enough to supply the maximum quantity of water that can be evaporated in the boiler. At the Centennial Exhibition a standard of 30 pounds per horse-power per hour was adopted and while this is a safe figure to use when calculating the size of boiler required for a steam-engine, it is too low to be considered as a basis for selecting feed-pumps, as the hereinafter considerations will show.

It is general practice among builders to furnish about 12 square feet of heating surface per horse-power, and it has been found that but little decrease of economy will take place if the boiler is forced to evaporate 4 pounds per hour per square foot of heating surface, instead of the 2½ pounds called for by the Centennial standard.

The A.S.M.E. committee on "Trial of Steam Boilers," in 1884 reported as its opinion that a boiler

[1] Contributed to Power by F. F. Nickel.

should be capable of developing its rated horse-power with easy firing, moderate draft and ordinary fuel, and further that it should be capable of delivering at least one-third more than its rated power to meet emergencies.

These considerations led to the adoption of 45 pounds per horse-power per hour as the quantity for which a boiler feed-pump should be calculated. This quantity must be delivered to the boiler at moderate speed, so that in case of low-water level in the boiler the pump can be speeded and the deficiency made up promptly. It is therefore good practice to reduce the speed of the boiler feed-pump to one-half of what the pump would be rated at for regular service.

The accompanying table gives average sizes of feed-pumps as furnished by the various builders, together with the proper speed, capacity and horse-power of the boilers they are intended to supply.

BOILER FEED-PUMPS

Size	Gallons per Revolution	Strokes per Minute	Piston Speed, Ft. per Min.	Gallons per Minute.	Pounds per Hour	Horse-power of Boiler Supplied at 45-lb. Rate
3 × 2 × 3	0.155	80	20	6.2	3,100	70
$3\frac{1}{2}$ × $2\frac{1}{4}$ × 4	0.265	76	25	10	5,000	110
$4\frac{1}{2}$ × $2\frac{3}{4}$ × 4	0.395	76	25	15	7,500	170
$5\frac{1}{4}$ × $3\frac{1}{2}$ × 5	0.805	70	29	28	14,000	310
6 × 4 × 6	1.265	66	33	42	21,000	470
$7\frac{1}{2}$ × $4\frac{1}{2}$ × 6	1.6	66	33	53	25,150	560
$7\frac{1}{2}$ × $4\frac{1}{2}$ × 8	2.15	60	40	65	32,500	720
$7\frac{1}{2}$ × $4\frac{1}{2}$ × 10	2.66	54	45	72	36,000	800
8 × 5 × 10	3.25	54	45	88	44,000	1000
9 × $5\frac{1}{4}$ × 10	3.6	54	45	97	48,500	1100
10 × 6 × 10	4.75	54	45	128	64,000	1400
12 × 7 × 10	6.45	54	45	174	87,000	1900
12 × 7 × 12	7.75	48	48	186	93,000	2100
14 × $8\frac{1}{2}$ × 10	9.55	54	45	258	129,000	2900
14 × $8\frac{1}{2}$ × 12	11.5	48	48	276	138,000	3100
16 × $10\frac{1}{4}$ × 10	14.0	54	45	378	189,000	4200
16 × $10\frac{1}{4}$ × 12	16.7	48	48	401	200,500	4500
$18\frac{1}{2}$ × 12 × 10	19.2	54	45	518	259,000	5800
$18\frac{1}{2}$ × 12 × 12	23.0	48	48	552	276,000	6100
20 × 14 × 10	26.4	54	45	713	356,500	8000
20 × 14 × 12	31.5	48	48	756	378,000	8400

X

HORSE-POWER OF PUMP[1]

A PUMP in doing a certain amount of work is known to consume 5 horse-power. The pump is $1\frac{1}{2} \times 6$ inches, with 15 strokes. The water is discharged into a reservoir, and the work of pumping the water through the pipe line requires 5 horse-power. Required, the pressure per square inch against which the plunger is pumping, all losses to be neglected.

The quantity of water which the pump is delivering must first be found. As the diameter of the plunger is $1\frac{1}{2}$ inches, the area is 1.767 square inches, and as the stroke is 6 inches and there are 15 strokes per minute, $1.767 \times 6 \times 15 = 159.03$ cubic inches of water pumped per minute.

If it were not known just what horse-power the pump was consuming, it could be found from the following formula:

Pounds of water pumped per minute $\times$ head in feet $\div$ 33,000.

But as the horse-power is known in this instance, $33,000 \times 5 \div$ pounds of water pumped per minute = head in feet. Since 1 pound of water contains 27.7 cubic inches, $159.03 \div 27.7 = 5.741$ pounds of water

[1] Contributed to Power by Frank L. Ferguson.

discharged per minute, so that by inserting this value in the formula we have: $33{,}000 \times 5 \div 5.741 = 28{,}740$ feet head, which the water is pumped against to consume 5 horse-power.

If it is desired to know the equivalent pressure per square inch acting against the pump plunger, all that is necessary is to multiply 28,740 by $0.434 = 12{,}473.16$ pounds per square inch pressure, as every foot-head equals a pressure of 0.434 pounds per square inch.

XI

TO INDICATE THE AMOUNT OF FEED-WATER PUMPED INTO BOILERS[1]

In no other part of the power plant is there more urgent need of some simple device for indicating the amount of work being done than with the apparatus for feeding water into the boilers. A counter for recording the number of pump strokes, a water meter, or other method of measuring the water, will show the amount pumped during a given period, but will not indicate the rate of flow at any instant. What is wanted is the equivalent of the ammeter on a switchboard. The gage glass cannot be said to be this equivalent, as it only shows the water level, not the rate at which the water is going in. An engineer who knows his plant can judge fairly well how much he is putting in by the speed of the pump, when that is in sight; but in many plants the practice of locating the feed-pumps in a separate room from the boiler room often renders even this unavailable.

It must be understood that these remarks apply only to the normal running conditions of the plant, as of course when tests are being taken suitable provision

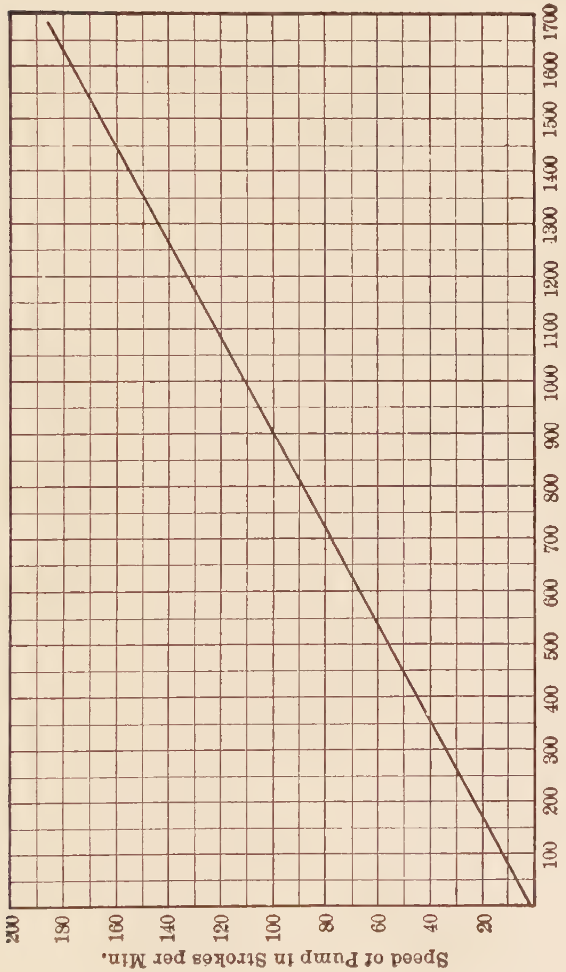
[1] Contributed to Power by F. Sanford.

must be made for accurately measuring the actual amount of feed-water used during a given period.

In a certain plant, consisting of six Babcock & Wilcox boilers, each capable of evaporating 12,000 pounds per hour, duplex steam-driven pumps were installed to supply the necessary feed-water. These were placed in the boiler room, and difficulty was experienced in keeping them in good condition on account of coal dust, which cut the pump plungers badly and caused excessive wear generally.

It was decided to install a motor-driven pump in an engine room at the rear of the boiler house, where it would be under the care of the engineer and free from the dust and grit. The controller was located in the boiler room. Current was obtained from the 3-wire system, making two voltages available, which allowed ample variation in the speed of the pump to meet the varying boiler loads. With this arrangement it was found necessary to provide means of indicating, so the boiler attendant could see whether the pump was running properly and at what speed.

The pump was first tested to ascertain the amounts of water delivered at various speeds against a head equal to the boiler pressure, and from these data a curve was plotted as shown in the accompanying diagram. A small 10-volt, shunt-wound generator, such as is used for gas-engine ignition, was connected to the pump and driven by a belt from the motor spindle. The shunt field was excited from the 125-volt direct-current mains, with a 32-candle-power lamp in series, thus obtaining a practically constant field strength.



Curve showing amounts of water in gallons delivered at various speeds

The voltage obtained from the small generator was then exactly proportional to the speed of the the pump.

The next step was to install a large illuminated-dial voltmeter directly above the controller, in which position it was visible from any part of the boiler room. Instead of showing volts, the voltmeter dial was marked to indicate pounds of water per minute, which were proportional to the speed of the pump, as shown by the curve. To check up the indications of the meter at any time, it was only necessary to take the pump speed and compare the meter reading with the corresponding amount of water shown on the curve. It was not necessary to take the efficiency of the pump into consideration, as the curve was plotted from the actual amount of water delivered at various speeds.

This device was found extremely useful to the boiler attendant, who, being able to read the amount he was pumping at any time, and knowing from long practice the variation of load on his boilers was able to anticipate the demand and keep the water level constant.

XII

PUMPING TAR AND OTHER HEAVY LIQUIDS

In many industries it is necessary to force heavy, viscous liquids through pipes. This involves difficulties not encountered in ordinary pumping, and requires machinery special in design and construction. When the liquid is heavy but not adhesive, as in the case of heavy oils, the action can be made fairly satisfactory and efficient by enlarging the valve openings, making the parts of the pump heavier and so arranging the passages of the pump that there is little liability of choking or clogging. When, however, the liquid is a fluid at high temperatures and a gelatinous adhesive paste or a rubbery solid, clinging to all surfaces and choking openings through which it should pass, as the temperature is lowered, a design differing materially from the ordinary pump must be used.

Tar, molasses and cocoa liquor present more obstacles to pumping than any other substances which it has been found feasible to move in this manner. Each of these liquids thickens into an almost solid mass when cold, rendering it very difficult to start the pump, if some special provision is not made and ample power provided. Another action which must be taken into account is the contraction of the area of the passages

and valves as the liquid cools and the consequent throttling which interferes with the liquid's passage and which the pump is forced to overcome. The skin friction of a liquid of this kind creates heat enough to partially alleviate this tendency to throttling when the velocity of the substance is maintained above a certain point and the pipe is not in such a position that the surrounding air will lower the temperature of the liquid below the solidifying point. Although not a common practice, it is well to lag all exposed piping used for conveying heavy oils or other substances of a similar nature.

Gas tar has a number of characteristics rendering it exceptionally difficult to pump. Its condition varies from a solid to a penetrating fluid within a small range of temperature. Two pumps which have proved very efficient in lifting and forcing gas tar were installed a short time ago at the plant of the Maryland Steel Company, of Sparrows Point, Maryland. They are of the standard triplex type, fitted with ball valves peculiarly adapted to this service. The exclusive use of gate valves in the piping system is also interesting. A very flexible power connection is obtained by the use of the Renolds silent chain and a 4-pole alternating-current 3-horse-power motor. The gearing consists of an 18-tooth pinion running at 950 revolutions per minute and a 120-tooth wheel running at 142 revolutions per minute. The chain used has links $\frac{3}{4}$-inch wide and $1\frac{1}{2}$ inches long. It transmits the 3-horse-power generated at 950 feet per minute, giving an excellent efficiency when the service is considered.

A liquid peculiarly difficult to handle is oil-refinery tar, which is usually very hot when it reaches the pump. There is a large percentage of suspended particles of various sizes present in this tar and also a certain amount of unrefined paraffine. The tar is sometimes heated to a temperature of 300 degrees; but quickly cools off if not properly handled, and coats the retaining valves and walls with layers of an adhesive substance closely resembling finely divided particles of coke. To overcome the difficulties the ordinary pump arrangement and design is materially changed.

A special pump for handling oil-refinery tar at the works of the Atlantic Refining Company, in Philadelphia has been designed. By a new arrangement, exceptionally large valve areas are made available, the valves being designed to permit the passage of the substance pumped with the least possible frictional resistance. The suction, discharge and pulsation chambers can be taken apart without unnecessary expenditure of time or labor, and each is in a position where it can be readily reached for cleaning. The pump is of the triplex type, and is fitted with ball valves, which, through test, have proved best adapted for the passage of heavy substances. There are a number of large hand-holes for cleaning the valves.

Machinery which will pump these adhesive oils and other similar substances can be used in many industries, and will save the laborious processes by which this class of work is generally accomplished.

XIII

PUMPING MACHINERY PERFORMANCES

The table which is on the opposite page was compiled by a prominent engineer in studying the work of municipal pumping machinery, and is a table giving the performances of twenty of the best known and most efficient pumping engines.

The vertical compound engine (No. 2) is notable as holding the record for a compound engine, having shown, on a 6-day test, a duty of 148,655,000 foot-pounds, with a steam consumption of 12.15 pounds per indicated horse-power hour, and having given an average annual duty of about 120,000,000 foot-pounds per 100 pounds of coal, a performance equaling that of many triple-expansion plants.

The Nordberg quadruple-expansion engine (No. 7 in the table) is notable for having established the record for low heat consumption, the figures being 162,132,500 foot-pounds per 1,000,000 B.T.U., or 186 B.T.U. per indicated horse-power per minute, the thermal efficiency being about 22.8 per cent.

The duties usually guaranteed per 1,000 pounds of dry steam are about 60,000,000 for compound condensing, 90,000,000 for triple-expansion condensing, and 110,000,000 for compounds with high-duty attachments, and 130,000,000 for triple-expansion machines with high-duty attachments.

PERFORMANCES OF PUMPING ENGINES, 1893–1903

The engines are of the Triple type, with two exceptions; on line 2 (E. D. Leavitt,) which is Compound; and on line 7 (Nordberg Mfg. Co.) which is Quadruple.

	Builder or Designer	Date of Test	Location of Engine	Capacity, in gallons per 24 hours. (In millions)	Piston Speed, in feet	Head: Feet	Head: Pounds	Steam Pressure, in lbs	Indicated Horse power	Percentage of Mechanical Efficiency	Steam, I. H. P., hour	B. T. U., I. H. P., min.	Duty, in foot-pounds per 1,000,000 B. T. U.	Duty, in foot-pounds per 1,000 pounds Steam	Percentage of Thermal Efficiency
1	Edward P. Allis Co.	1893	Milwaukee, Wis.	18	203	162.6	70.4	121	574	91	11.68	217.6	137,656,000	154,048,700	19.40
2	E. D. Leavitt	1894	Louisville, Ky	16	371	193.3	83.7	137	643	93	12.15	222.5	137,565,000	148,655,000	19.07
3	E. D. Leavitt	1895	Boston, Mass.	20	607	137.2	59.4	176	576	89.5	[1]11.22	[1]204.3	[1]144,500,000	[1]157,843,000	20.76
4	Lake Erie Eng. Wks.	1897	Buffalo, N. Y.	30	208	199	86	167	1,186	95	12.39	231.2	135,000,000	152,000,000	18.35
5	Snow St. Pump Co. .	1898	Indianapolis, Ind....	20	215	205	89	156	776	95.4	11.26	207.6	150,000,000	167,800,000	20.45
6	Edward P. Allis Co..	1899	Cleveland, Ohio.....	20	200	202	87.5	149	770	93.7	11.45	213	145,220,000	161,990,000	19.9
7	Nordberg Mfg. Co. .	1899	Wildwood, Pa	6	256	604	262	200	712	93	12.26	186	162,132,500	149,500,000	22.8
8	Edward P. Allis Co..	1899	St.Louis, Mo.,No. 7.	10	175	294	127.5	136	549	94.3	11.63	216.8	143,404,000	160,455,000	18.44
9	Edward P. Allis Co..	1899	St. Louis, Mo.,No.8.	10	175	294	127.5	137	545	94.9	11.63	216.5	144,365,000	161,530,000	18.59
10	Edward P. Allis Co..	1900	Hackensack, N. J....	12	211	267	115	173	603	94	11.05	211.9	146,403,400	168,532,800	20.00
11	Edward P. Allis Co..	1900	St. Louis, Mo., No. 9.	15	198	293	127	127	813	96	10.78	204.4	155,237,450	176,419,600	20.78
12	Edward P. Allis Co..	1900	St. Louis, Mo., No. 10	15	197	292	126	126	802	96.8	10.68	202	158,077,320	179,454,250	21.00
13	Edward P. Allis Co..	1900	Boston, Mass.	30	195	140	61	185	748	93.3	10.33	196	163,925,300	178,497,000	21.63
14	E. D.Leavitt	1901	Cambridge, Mass....	21	496	190	82	178	780	93.4	12.17	221	135,816,000	146,173,000	19.43
15	Holly Mfg. Co	1901	Boston, Mass.	35	300	45	19.5	151	323	88	11.10	210	141,532,000	157,349,000	20.50
16	Holly Mfg. Co	1901	Boston (Spot Pond)	20	248	125	54	150	464	96.5	11.01	203.4	156,592,000	173,620,000	20.85
17	E. D. Leavitt	1903	New Bedford, Mass .	10	480	185	80	181	342	95	13.46	223.9	150,010,000	140,000,000	18.95
18	Edward P. Allis Co..	1903	St Louis, Mo.,No.11	15	197	293	126.9	138	800	97.2	10.84	204.8	155,800,000	177,300,000	20.72
19	Edward P. Allis Co..	1903	St.Louis, Mo.,No.12	15	197	293	126.9	138	796	97.7	10.88	205.2	156,900,000	177,200,000	20.67
20	Allis-Chalmers Co. .	1904	St. Louis, Mo. No 1..	20	198	240	104	...	...	...	....	....	....	176,866,000	... ..

[1] Exclusive of auxiliaries.

XIV

GENERAL DIRECTIONS FOR SETTING UP AND OPERATING PUMPS

In setting up a pump the first requisite is to provide for a full and steady supply of water or other fluid. To accomplish this observe carefully the following points.

The *suction pipe* sizes are given by various pump manufacturers in tables or upon application, and in no case should the size of the pipe be reduced to less size than the manufacturers give. In some cases where the suction pipe is long it must be larger than the size given to overcome friction. Make the pipe connections as short as possible with the fewest number of bends possible and these as easy (long radius) as possible.

In laying *suction pipe*, a *uniform grade* should be maintained, thereby avoiding air pockets or summits. Grade the suction pipe toward the supply, with a drop of not less than 6 *inches in each* 100 *feet.* It will be found economical to have grade given by a civil engineer.

The *suction pipe* and its connections *must be tight,* as a very small leak will supply the pump with air to its full capacity so that little or no water will be obtained,

according to the size of the leak. Before covering the suction pipe it is recommended that it be tested with a pressure of not less than 25 or more than 50 pounds per square inch, to discover any leaks.

Wrought-iron pipe may be used for suction pipe of small sizes, but cast-iron flanged pipe is recommended for all sizes in which it can be obtained. When bell and spigot pipe is used it should be laid with the direction of the current from the bell end toward the spigot end.

All valves in suction and discharge pipe should be *gate valves*.

A *suction air chamber* is an advantage on long or high suctions, and is particularly recommended for single pumps, on all fire pumps, and any pumps which are to run at high speed, expecially for pumps of short stroke.

A *foot-valve*, under these conditions, insures a quick starting of the pump by maintaining the pipe full of water and free from air. When a foot-valve is used see that the area of its valve-seat openings is not less than the area of the pipe.

A *strainer* is always desirable but not necessary when water is clear and free from foreign matter that will clog the valves and passages of the pump. The area of the strainer openings should be at least four times the area of the pipe, to equalize the friction of water through the small openings, and because some of them are liable to become clogged. When strainers are used they must be frequently inspected and cleaned.

Extreme caution must be exercised while pipe is being laid and pump connected, to prevent foreign matter, such as sticks, waste, and rubbish from entering the pipe. Chips from threading pipe, sand, etc., will quickly cut the cylinder, piston, and valve of a pump, doing more damage than years of proper use, or perhaps, entirely disabling it.

A *priming pipe* connected to a supply above the pump or under pressure is a convenience for quick starting, and a necessity for a fire pump, and most large pumps of all classes.

Hot water cannot be raised to any considerable hight by suction. Thick liquids and hot water should always flow to the pump by gravitation.

Steam and exhaust pipes should be as straight as possible and of the full size called for by manufacturer's tables.

In connecting the steam-pipe, proper allowance should be made for expansion. A gate throttle valve should be placed in the steam-pipe close to the pump. Means should be provided for draining this pipe before starting the pump.

A *heater* may be placed in the exhaust pipe to advantage.

To prevent freezing, drain the pump by opening all cocks and plugs provided for the purpose. In piping from these drips, valves should be placed close to the pump cylinders. The steam and water cylinder drips should never be connected into the same pipe unless a check valve is placed so as to close towards the water cylinder to keep it free from steam.

Erecting of a pump should be done by a thoroughly competent man.

Foundations suitable for the pump should always be provided.

All pipes should be properly supported so as to relieve the pump flanges from undue strains.

Keep the *steam cylinder* well oiled, especially just before stopping.

Keep the *stuffing-boxes* well and evenly filled with a good quality of packing. Don't screw them too tight.

Let the steam end alone if the pump begins to run badly, until fully satisfied that there is no obstruction in the water cylinder, water valves or pipes.

The pump should be located, if possible, in a light, dry, warm and clean place and have good care. Do not overlook the importance of this last suggestion.

Do not pull the pump apart to see what is inside as long as it does its work well.

XV

USEFUL INFORMATION

WATER

ONE cubic inch weighs .0361 pounds.

One pound = 27.7 cubic inches.

One cubic foot = 62.4245 pounds at 39 degrees Fahrenheit; 7.48 gallons U. S.; 6.2321 gallons imperial.

One gallon U. S. = 8.33111 pounds; 231 cubic inches .13368 cubic feet.

One imperial gallon = 10 pounds at 62 degrees Fahrenheit; 277.274 cubic inches; .16046 cubic feet.

One pound pressure = 2.31 feet in hight.

One foot in hight — .433 pounds pressure.

Petroleum weighs 6½ pounds per U. S. gallon, 42 gallons to the barrel.

To convert imperial gallons into U. S. gallons, multiply by the factor 1.2. To convert U. S. gallons into imperial gallons, multiply by the factor .8333.

A miner's inch is a measure for flow of water, and is the quantity of water that will flow in one minute through an opening 1 inch square in a plank 2 inches thick under a head of 62 inches to the center of the orifice. This is equivalent, approximately, to 1.53 cubic feet, or 11½ gallons per minute.

To find the diameter of pump plungers to pump a

given quantity of water at 100 feet piston speed per minute, divide the number of gallons by 4, then extract the square root, and the result will be the diameter in inches of the plungers.

To find the number of gallons delivered per minute by a single double-acting pump at 100 feet piston speed per minute, square the diameters of the plungers, then multiply by 4.

To find the horse-power necessary to elevate water to a given hight, multiply the weight of the water elevated per minute by the hight in feet and divide the product by 33,000 (an allowance should be made for water friction and a further allowance for losses in the steam cylinder, say from 20 to 30 per cent.).

The mean pressure of the atmosphere is usually estimated at 14.7 pounds per square inch, so that with a perfect vacuum it will sustain a column of mercury 29.9 inches, or a column of water 33.9 feet high at sea level.

To determine the proportion between the steam and pump cylinder, multiply the given area of the pump cylinder by the resistance on the pump in pounds per square inch, and divide the product by the available pressure of steam in pounds per square inch. The product equals the area of the steam cylinder. To this must be added an extra area to overcome the friction, which is usually taken at 25 per cent.

The resistance of friction in the flow of water through pipes of uniform diameter is independent of the pressure and increases *directly* as the length and the square of the velocity of the flow, and *inversely* as the diameter

of the pipe. With wooden pipes the friction is 1.75 times greater than in metallic. Doubling the diameter increases the capacity four times.

To determine the velocity in feet per minute necessary to discharge a given volume of water in a given time, multiply the number of cubic feet of water by 144 and divide the product by the area of the pipe in inches.

To determine the area of a required pipe, the volume and velocity of water being given, multiply the number of cubic feet of water by 144 and divide the product by the velocity in feet per minute.

XVI

USEFUL TABLES

Hights in Feet to Which Pumps will Elevate Water

Steam Pressure, 50 Pounds per Square Inch at the Pump
No Allowance made for Friction in Pipes, etc.

Diameter of Steam Cylinders	DIAMETER OF WATER CYLINDERS																
	2 Inch	2¼ Inch	3 Inch	3½ Inch	4 Inch	5 Inch	6 Inch	7 Inch	8 Inch	9 Inch	10 Inch	10½ Inch	12 Inch	14 Inch	16 Inch	18 Inch	20 Inch
3½	230	147	102	75	58	37											
4	300	192	134	134	75	48	34										
5	469	300	209	153	117	75	52	38									
6	675	432	300	221	169	108	75	55	42	33							
7	920	588	408	300	230	147	102	75	57	45	37						
8	...	768	533	344	300	192	141	98	75	59	48	44					
9	...	972	675	496	380	243	169	124	95	75	61	55	42				
10	...	...	833	612	469	300	208	153	117	94	75	68	50	38			
12	...	...	...	881	675	432	300	220	169	133	108	97	75	55	42		
14	...	...	...	...	920	588	408	300	228	182	147	133	102	75	57	45	
16	...	...	...	...	...	768	564	392	300	236	192	174	141	98	75	59	48
18	...	...	...	...	...	972	650	490	379	300	243	220	162	122	95	75	61
20	...	...	...	...	...	...	833	600	469	370	300	272	208	150	117	92	75
22	...	...	...	...	...	...	1008	741	567	448	364	329	252	185	142	112	91
24	...	...	...	...	...	...	...	882	675	533	432	392	300	220	169	133	108
26	...	...	...	...	...	...	....	1034	788	626	508	460	356	258	197	156	127
28	...	...	...	...	...	...	....	....	919	726	588	533	407	300	230	181	147
30	...	...	...	...	...	...	....	....	1054	834	676	612	468	345	263	208	169
32	...	...	...	...	...	...	....	....	....	948	798	697	533	391	300	237	192
34	...	...	...	...	...	...	....	....	....	1070	868	786	603	442	339	268	217
36	...	...	...	...	...	...	....	....	....	....	972	881	675	495	380	300	243

The maximum limit of piston speed depends upon the head pumped against.

Friction Loss in Pounds Pressure per Square Inch

For each 100 Feet of Length in Different Size Clean Iron Pipes Discharging given Quantities of Water per Minute

Gallons discharged per Minute	½ Inch	¾ Inch	1 Inch	1¼ Inch	1½ Inch	2 Inch	2½ Inch	3 Inch	3½ Inch	4 Inch	5 Inch	6 Inch
	Friction Loss in Pounds	Friction Loss in Pounds	Friction Loss in Pounds	Friction Loss in Pounds	Friction Loss in Pounds	Friction Loss in Pounds	Friction Loss in Pounds	Friction Loss in Pounds	Friction Loss in Pounds	Friction Loss in Pounds	Friction Loss in Pounds	Friction Loss in Pounds
5	24.6	3.3	.84	.31	.12	....	....	....	....	..	....	....
10	96.0	13.0	3.16	1.05	.47	.12	....	....	....	...	....	....
15	....	28.7	6.98	2.38	.97	....	....	....	....	...	....	....
20	....	50.4	12.3	4.07	1.66	.42	....	....	....	...	....	....
25	....	78.0	19.0	6.40	2.62	....	.21	.10	....	...	....	....
30	....	....	27.5	9.15	3.75	.91	....	....	....	....	....	....
35	....	....	37.0	12.04	5.05	....	....	....	....	....	....	....
40	....	....	48.0	16.1	6.52	1.60	....	....	....	....	....	....
45	....	....	....	20.2	8.15	....	....	....	....	....	....	....
50	....	....	....	24.9	10.0	2.44	.81	.35	.16	.09	.03	....
75	....	....	....	56.1	22.4	5.32	1.80	.74	.34	....	....	....
100	....	....	....	....	39.0	9.46	3.20	1.31	.60	.33	.12	.05
125	....	....	....	....	....	14.9	4.89	1.99	.90	....	....	....
150	....	....	....	....	....	21.2	7.0	2.85	1.32	.69	.25	.10
175	....	....	....	....	....	28.1	9.46	3.85	1.78	....	....	....
200	....	....	....	....	....	37.5	12.48	5.02	2.32	1.22	.42	.17
250	....	....	....	....	....	....	19.66	7.76	3.55	1.89	.65	.26
300	....	....	....	....	....	....	28.06	11.2	5.23	2.66	.93	.37
350	....	....	....	....	....	....	....	15.2	7.0	3.65	1.28	.50
400	....	....	....	....	....	....	....	19.5	9.0	4.73	1.68	.65
450	....	....	....	....	....	....	....	25.0	11.60	6.03	2.10	.81
500	....	....	....	....	....	....	....	30.8	14.26	7.41	2.70	.96

Weight and Capacity of Different Standard Gallons of Water

	Cubic Inches in a Gallon	Weight of a Gallon in pounds	Gallons in a Cubic Foot	Weight of a cubic foot of water, English standard, 62.321 pounds Avoirdupois.
Imperial or English ...	277.274	10.00	6.232102	
United States	231.	8.33111	7.480519	

Weight of Crude Petroleum, $6\frac{1}{2}$ pounds per U. S. gallon, } 42 gallons to the barrel.
Weight of Refined Petroleum, $6\frac{1}{2}$ pounds per U. S. gallon, }
A "miner's inch" of water is approximately equal to a supply of 12 U. S. gallons per minute.

Pounds Pressure Lost by Friction

In each 100 Feet of $2\frac{1}{2}$-inch Fire Hose, for given Discharges of Water per Minute

Diameter of Nozzle, Inches	PRESSURE AT HOSE NOZZLE									
	Head in pounds per sq. in.	20	30	40	50	60	70	80	90	100
	Head in feet	46.2	69.3	92.4	115.5	138.6	161.7	184.8	207.9	231.0
1	Gallons discharged...	110	134	155	173	189	205	219	232	245
	Rubber hose, pounds .	4.35	6.40	8.40	10.20	12.80	14.80	17.0	19.20	20.50
	Leather hose, pounds.	6.33	8.53	10.83	13.10	15.34	17.79	20.11	22.40	24.83
$1\frac{1}{8}$	Gallons discharged ..	139	170	196	219	240	259	277	294	310
	Rubber hose, pounds .	6.79	10.16	13.60	17.05	20.59	24.0	27.0	30.0	33.0
	Leather hose, pounds.	9.05	12.71	16.38	20.11	23.88	27.61	31.41	35.24	39.07
$1\frac{1}{4}$	Gallons discharged ..	171	210	242	271	297	320	342	363	383
	Rubber hose, pounds .	10.28	15.64	20.85	25.46	29.50	39.0	43.81	49.42	55.0
	Leather hose, pounds.	12.84	19.0	24.07	30.11	35.94	41.57	47.36	53.25	59.20
$1\frac{3}{8}$	Gallons discharged ..	207	253	293	327	358	387	413	439	462
	Rubber hose, pounds .	15.0	22.96	29.40	40.50	48.20	55.70	64.70	72.0	79.26
	Leather hose, pounds.	18.81	26.39	35.01	43.38	52.0	60.40	68.59	76.73	84.87

HORIZONTAL AND VERTICAL DISTANCES REACHED BY JETS

Diameter of Nozzle, Inches	PRESSURE AT NOZZLE									
	Head in pounds, per sq. in.	20	30	40	50	60	70	80	90	100
	Head in feet	40.2	60.3	92.4	115.5	138.0	161.7	184.8	207.9	231.0
1	Gallons discharged	110	134	155	173	180	205	219	232	245
	Horizontal distance of jet	70	00	100	120	142	150	168	178	186
	Vertical distance of jet	43	62	79	94	108	121	131	140	148
1⅛	Gallons discharged	131	170	106	219	240	250	277	294	310
	Horizontal distance of jet	71	03	113	132	148	103	175	186	193
	Vertical distance of jet	43	03	81	97	112	125	137	148	157
1¼	Gallons discharged	171	210	242	271	207	320	342	363	383
	Horizontal distance of jet	73	06	118	138	150	172	186	198	207
	Vertical distance of jet	43	03	82	99	115	120	142	154	164
1⅜	Gallons discharged	207	253	203	327	358	387	413	439	402
	Horizontal distance of jet	75	100	124	140	100	184	200	213	224
	Vertical distance of jet	44	05	85	102	118	133	146	158	169

Pressure of Water

Feet Head	Pressure per Square Inch	Feet Head	Pressure per Square Inch	Feet Head	Pressure per Square Inch	Feet Head	Pressure per Square Inch	Feet Head	Pressure per Square Inch	Feet Head	Pressure per Square Inch
1	0.43	32	13.86	63	27.29	94	40.72	225	97.46	385	166.78
2	0.86	33	14.29	64	27.72	95	41.15	230	99.63	390	168.94
3	1.30	34	14.72	65	28.15	96	41.58	235	101.79	395	171.11
4	1.73	35	15.16	66	28.58	97	42.01	240	103.96	400	173.27
5	2.16	36	15.59	67	29.02	98	42.45	245	106.13	425	184.10
6	2.59	37	16.02	68	29.45	99	42.88	250	108.29	450	195.00
7	3.03	38	16.45	69	29.88	100	43.31	255	110.46	475	205.77
8	3.46	39	16.89	70	30.32	105	45.48	260	112.62	500	216.58
9	3.89	40	17.32	71	30.75	110	47.64	270	116.96	525	227.42
10	4.33	41	17.75	72	31.18	115	49.81	275	119.12	550	238.25
11	4.76	42	18.19	73	31.62	120	51.98	280	121.29	575	249.09
12	5.20	43	18.62	74	32.05	125	54.15	285	123.45	600	259.90
13	5.63	44	19.05	75	32.48	130	56.31	290	125.62	625	270.73
14	6.06	45	19.49	76	32.92	135	58.48	295	127.78	650	281.56
15	6.49	46	19.92	77	33.35	140	60.64	300	129.95	675	292.40
16	6.93	47	20.35	78	33.78	145	62.81	305	132.12	700	303.22
17	7.36	48	20.79	79	34.21	150	64.97	310	134.28	725	314.05
18	7.79	49	21.22	80	34.65	155	67.14	315	136.46	750	324.88
19	8.22	50	21.65	81	35.08	160	69.31	320	138.62	775	335.72
20	8.66	51	22.09	82	35.52	165	71.47	325	140.79	800	346.54
21	9.09	52	22.52	83	35.95	170	73.64	330	142.95	825	357.37
22	9.53	53	22.95	84	36.39	175	75.80	335	145.12	850	368.20
23	9.96	54	23.39	85	36.82	180	77.97	340	147.28	875	379.03
24	10.39	55	23.82	86	37.25	185	80.14	345	149.45	900	389.86
25	10.82	56	24.26	87	37.68	190	82.30	350	151.61	925	400.70
26	11.26	57	24.69	88	38.12	195	84.47	355	153.78	950	411.54
27	11.69	58	25.12	89	38.35	200	86.63	360	155.94	975	422.35
28	12.12	59	25.55	90	38.93	205	88.80	365	158.10	1000	433.18
29	12.55	60	25.99	91	39.42	210	90.96	370	160.27	1500	649.70
30	12.99	61	26.42	92	39.85	215	93.13	375	162.45	2000	866.30
31	13.42	62	26.85	93	40.28	220	95.30	380	164.61	3000	1299.50

Areas of Circles, Advancing by Eighths

Diam.	AREAS 0	1/8	1/4	3/8	1/2	5/8	3/4	7/8
0	.0	.0123	.0491	.1105	.1964	.3068	.4418	.6013
1	.7854	.9940	1.2272	1.4849	1.7671	2.0739	2.4053	2.7612
2	3.14	3.55	3.98	4.43	4.91	5.41	5.94	6.49
3	7.07	7.67	8.30	8.95	9.62	10.32	11.05	11.79
4	12.57	13.36	14.19	15.03	15.90	16.80	17 72	18.67
5	19.64	20.63	21.65	22.69	23.76	24.85	25.97	27.11
6	28.27	29.47	30.68	31.92	33.18	34.47	35.79	37.12
7	38.49	39.87	41.28	42.72	44.18	45.66	47.17	48.71
8	50.27	51.85	53.46	55.09	56.75	58.43	60.13	61.86
9	63.62	65.40	67.20	69.03	70.88	72.76	74.66	76.59
10	78.54	80.52	82.52	84.54	86.59	88.66	90.76	92.89
11	95.03	97.21	99.40	101.62	103.87	106.14	108.43	110.75
12	113.10	115.47	117.86	120.28	122.72	125.19	127.68	130.19
13	132.73	135.30	137.89	140.50	143.14	145.80	148.49	151.20
14	153.94	156.70	159.48	162.30	165.13	167.99	170.87	173.78
15	176.71	179.67	182.65	185.66	188.69	191.75	194.83	197.93
16	201.06	204.22	207.39	210.60	213.82	217.08	220.35	223.65
17	226.98	230.33	233.71	237.10	240.53	243.98	247.45	250.95
18	254.47	258.02	261.59	265.18	268.80	272.45	276.12	279.81
19	283.53	287.27	291.04	294.83	298.65	302.49	306.35	310.24
20	314.16	318.10	322.06	326.05	330.06	334.10	338.16	342.25
21	346.36	350.50	354.66	358.84	363.05	367.28	371.54	375.83
22	380.13	384.46	388.82	393.20	397.61	402.04	406.49	410.97
23	415.48	420.00	424.56	429.13	433.74	438.36	443.01	447.69
24	452.39	457.11	461.86	466.64	471.44	476.26	481.11	485.98
25	490.87	495.79	500.74	505.71	510.71	515.72	520.77	525.84
26	530.93	536.05	541.19	546.35	551.55	556.76	562.00	567.27
27	572.56	577.87	583.21	588.57	593.96	599.37	604.81	610.27
28	615.75	621.26	626.80	632.36	637.94	643.55	649.18	654.84
29	660.52	666.23	671.96	677.71	683.49	689.30	695.13	700.98
30	706.86	712.76	718.69	724.64	730.62	736.62	742.64	748.69
31	754.77	760.87	766.99	773.14	779.31	785.51	791.73	797.98
32	804.25	810.54	816.86	823.21	829.58	835.97	842.39	848.83
33	855.30	861.79	868.31	874.85	881.41	888.00	894.62	901.26
34	907.92	914.61	921.32	928.06	934.82	941.61	948.42	955.25
35	962.11	969.00	975.91	982.84	088.80	996.78	1003.8	1010.6

AREAS OF CIRCLES, ADVANCING BY EIGHTHS

	AREAS							
Diam.	0	1/8	1/4	3/8	1/2	5/8	3/4	7/8
36	1017.9	1025.0	1032.1	1039.2	1046.3	1053.5	1060.7	1068.0
37	1075.2	1082.5	1089.8	1097.1	1104.5	1111.8	1119.2	1126.7
38	1134.1	1141.6	1149.1	1156.6	1164.2	1171.7	1179.3	1186.9
39	1194.6	1202.3	1210.0	1217.7	1225.4	1233.2	1241.0	1248.8
40	1256.6	1264.5	1272.4	1280.3	1288.2	1296.2	1304.2	1312.2
41	1320.3	1328.3	1336.4	1344.5	1352.7	1360.8	1369.0	1377.2
42	1385.4	1393.7	1402.0	1410.3	1418.6	1427.0	1435.4	1443.8
43	1452.2	1460.7	1469.1	1477.6	1486.2	1494.7	1503.3	1511.9
44	1520.5	1529.2	1537.9	1546.6	1555.3	1564.0	1572.8	1581.6
45	1590.4	1599.3	1608.2	1617.0	1626.0	1634.9	1643.9	1652.9
46	1661.9	1670.9	1680.0	1689.1	1698.2	1707.4	1716.5	1725.7
47	1734.9	1744.2	1753.5	1762.7	1772.1	1781.4	1790.8	1800.1
48	1809.6	1819.0	1828.5	1837.9	1847.5	1857.0	1866.5	1876.1
49	1885.7	1895.4	1905.0	1914.7	1924.4	1934.2	1943.9	1953.7
50	1963.5	1973.3	1983.2	1993.1	2003.0	2012.9	2022.8	2032.8

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